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Steam Engines and Pump

147 Illustrations

By

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ENGINE MANAGEMENT
ENGINE INSTALLATION
PUMPS

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PREFACE

The volumes of the International Library of Technology are made up of Instruction Papers, or Sections, comprising the various courses of instruction for students of the International Correspondence Schools. The original manuscripts are prepared by persons thoroughly qualified both technically and by experience to write with authority, and in many cases they are regularly employed elsewhere in practical work as experts. The manuscripts are then carefully edited to make them suitable for correspondence instruction. The Instruction Papers are written clearly and in the simplest language possible, so as to make them readily understood by all students. Necessary technical expressions are clearly explained when introduced.

The great majority of our students wish to prepare themselves for advancement in their vocations or to qualify for more congenial occupations. Usually they are employed and able to devote only a few hours a day to study. Therefore every effort must be made to give them practical and accurate information in clear and concise form and to make this information include all of the essentials but none of the non-essentials. To make the text clear, illustrations are used freely. These illustrations are especially made by our own Illustrating Department in order to adapt them fully to the requirements of the text.

In the table of contents that immediately follows are given the titles of the Sections included in this volume, and under each title are listed the main topics discussed.

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CONTENTS

NOTE.—This volume is made up of a number of separate Sections, the page numbers of which usually begin with 1. To enable the reader to distinguish between the different Sections, each one is designated by a number preceded by a Section mark (§), which appears at the top of each page, opposite the page number. In this list of contents, the Section number is given following the title of the Section, and under each title appears a full synopsis of the subjects treated. This table of contents will enable the reader to find readily any topic covered.

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ENGINE MANAGEMENT

(PART 1.)

TAKING CHARGE.

1. The first duty of an engineer in taking charge of a power plant is to make a thorough inspection of all parts of it in order to become familiar with every detail of the engines, boilers, pumps, and their appurtenances. He should notice particularly if any parts of the engines, pumps, etc., such as cylinder heads, valve-chest covers, pump or condenser bonnets, connecting or eccentric rods, are disconnected or removed. If any parts have been removed, advantage should be taken of the opportunity to examine them. The condition of the interior parts of the engines should be noted. If the cylinder heads are off, he should examine and try with a wrench the follower bolts and piston-rod nuts. He should look carefully at the walls of the cylinders, and if they are cut, grooved, or pitted, he should make a note of the fact for future reference and comparison. The clearance between cylinder head and piston should be measured and marked on the guides with a center punch in order to ascertain and preserve a record of the space there is to spare for taking up the wear on connecting-rod journals. The cylinders should be wiped out thoroughly with oily waste and a liberal coating of cylinder oil should be applied to the wearing surfaces before closing up the cylinders.

2. If the valve-chest covers are off, the condition of the valves and seats should be noted. The lead of the valves should be examined; this may be done by turning the engines over by hand, having first poured a little oil into the journals. The valve gear should be watched during the operation of turning the engine, in order to discover if there is any derangement of the valve gear; if there is any obstruction to the engine turning freely, it will be revealed during the turning process.

3. If the pump-valve bonnets are off, the valves and valve chambers should be examined. If the valves are of hard, flat rubber and any of them are much worn on their lower faces, they should be turned over; if they are curled from standing dry for some time, or other causes, they should be faced off.

4. If the condenser bonnets are off, examine the interior of the condenser in order to ascertain its condition. If a surface condenser, fill the steam side of the condenser with water and look for leaky tubes; replace with new ones any tubes that may be split and renew all leaky tube packing.

5. Just before replacing any of the covers, bonnets, or cylinder heads, a final examination should be made to see that no tools, waste, or other foreign matter are left inside; look particularly for monkeywrenches, hammers, cold chisels, hand lamps, and pieces of waste, as it is quite a usual occurrence to find one or more of these articles inside of the machinery, left there by careless workmen; it is advisable that the engineer in charge should attend to this duty personally. Prior to replacing covers, bonnets, or cylinder heads, the gaskets should be examined; if they are torn or worn out, new ones should be used, and a thin coating of black lead (graphite) should be applied to them before they are put in place.

6. If the connecting-rod is disconnected at the crankpin end, an excellent opportunity to examine the condition of

the crankpin and brasses presents itself; if they are cut or are rough, they should be scraped down with a scraper and finished off with a smooth file, but it must be skilfully done; in connecting up, the crankpin brasses should not be set up too tight upon the pin; better leave them a little slack, to be taken up after the engine has been running for awhile.

7. Examine the piston rod and valve-stem stuffingboxes and put a turn or more of packing in them, as may be required. Examine the cylinder relief valves, if any are fitted to the engine; also examine the drain cocks to see that they are not stopped up.

8. Having put the engine in good order and gotten it ready for steam, the engineer should turn his attention to the feedwater apparatus; he should examine the main feed-pump most carefully. If it is a plunger pump, note the condition of the plunger packing and repack the stuffing-box if necessary; if it is a piston pump, examine the piston-rod packing and put in a turn or two of packing if the box will take it. Examine the petcock and see that it is not stopped up.

9. When satisfied that the pump is in good order, proceed to trace up the pipes. Commence with the suction pipe at the pump; follow it up and examine every foot of it to and from the filter, feedwater heater, or grease extractor, if there be one or the other, to the source of the feedwater supply, wherever that may be, and make sure that there is nothing the matter with this supply. Now start at the pump again and follow up the delivery pipe, tracing it through all its windings, noting every bend and connection, if any; also note where it enters and leaves the feedwater heater, purifier, or economizer, as the case may be, then on through the check-valve and globe valve to the point where it enters the boiler; take out the check-valve and carefully examine it, as well as its seat; wipe off the valve and valve seat with oily waste, and if it is in good condition, replace

the valve, but if found in poor order, repair it or replace it with a new one immediately. Try the globe valve in the feedpipe; if it works stiffly, oil the stem and thread and run the valve up and down until it works freely. Treat all globe valves in a similar manner, and if any of them need packing, attend to it at once. Trace up the auxiliary feedpipes, both suction and delivery, in the same manner and with the same care and attention that was given the main feedpipes.

10. Trace up, from beginning to end, all the auxiliary steam and exhaust piping to and from the various auxiliary engines and pumps, neglecting nothing. Note if the exhaust pipes of the auxiliary system lead to a condenser; if so, locate the valves for changing the auxiliary machinery from non-condensing to condensing and vice versa.

11. If the main engines are condensing, examine the air pump and circulating pump and their valves and trace up all steam pipes and water pipes leading to and from them, whether the pumps are operated by the main engine or independently.

12. Locate all cocks and valves in the piping and ascertain what every one of them is for; locate particularly all those connected with the feedwater supply system, both in the steam pipes and in the water pipes.

13. If the plant is a modern one and is supplied with the latest economical and other appliances and apparatus, such as a separator, grease extractor or filter, feedwater purifier, heater or economizer, evaporator, superheater, reheaters, etc., they must be included in the preliminary inspection and should receive the same care and attention as the other parts of the machinery.

14. In some plants the water of condensation from the steam pipes, valve chest, cylinders, steam jacket, and wherever else it may collect, is saved for use in the boilers;

this is accomplished by leading the various drain pipes into a manifold, from which a single pipe conveys the water to a steam trap, and from thence, by another pipe, to the hotwell or feed tank, where it mingles with the feedwater. These drain pipes should be traced up and examined from beginning to end. Look for leaky or broken joints and split pipes; if any are found, they must be repaired at once. Look also for badly rusted places in the pipes. Note if they are exposed to unusual dampness or dripping water; if such is the case, a coat of thick red-lead paint or paraffin varnish will afford considerable protection. Similar treatment should be accorded to any other iron or steel piping under floors or in places not easily accessible.

15. If during this inspection anything is found that is out of order, it should be repaired at once. Tools and stores for the ordinary repairs are generally provided.

STARTING AND STOPPING ENGINES.

INTRODUCTION.

16. Owing to the great variety of engines and valve gears in use and to the great difference in the sizes and power of steam plants, involving a wide range of appliances and apparatus, it is not possible to give specific directions in detail for starting and stopping each and every one of them. General instructions, with a few examples, can only be given, but it is the intention to make them full enough to enable the intelligent engineer, by using a little judgment and discretion, to apply them to all types of engines.

GENERAL INSTRUCTIONS.

17. Warming Up and Getting Ready.—The engine having previously been put in thorough order and the fires having been lighted in the boiler, it is assumed that the

steam pressure is now approaching its working point. About 15 or 20 minutes before starting the engine, raise the stop-valves just off their seats and let a little steam flow into the steam pipe; open the drain cock on the steam pipe just above the throttle. When the steam pipe is thoroughly warmed up and steam blows through the drain pipe, close the drain cock and open the throttle just enough to let a little steam flow into the valve chest and cylinder, or use the by-pass around the throttle, if one is fitted. Open the cylinder relief valves (or drain cocks), also the drain cocks on the valve chest and exhaust pipe, if a non-condensing engine. If cylinders are jacketed, turn the steam into the jackets and open the jacket drain cocks. While the engine is warming up, fill the oil cups and sight-feed lubricator. Squirt a little oil into all the small joints and journals that are not fitted with oil cups. Wipe off the guides with oily waste and squirt some oil over them. By this time the engine is getting warm; if fitted with by-pass valves, use them to admit steam into both ends of the cylinder. Further operations of warming up the cylinder will depend somewhat on the type of engine and valve gear; therefore, additional instructions regarding this matter will be given under their respective headings. In general, however, all cylinders, especially if they are large and intricate castings, should be warmed up slowly, as sudden and violent heating of a cylinder of this character is very liable to crack the casting by unequal expansion.

18. An excellent and economical plan for warming up the steam pipe and the engine is to open the stop-valves and throttle valve at the time or soon after the fires are lighted in the boilers, permitting the heated air from the boilers to circulate through the engine, thus warming it up gradually and avoiding the accumulation of a large quantity of water of condensation in the steam pipe and cylinder. When pressure shows on the boiler gauge or steam at the drain pipes of the engine, the stop-valves and throttle may be closed temporarily, but not hard down on their seats. When this

method of warming up the engine is adopted, the safety valves should not be opened while steam is being gotten up.

19. In attending to these preliminary arrangements certain precautions should be taken. For example, stop-valves and throttle valves should never be opened quickly or suddenly and thus permit a large volume of steam to flow into a cold steam pipe or cylinder. If this is done, the first steam that enters will be condensed and a partial vacuum will be formed. This will be closely followed by another rush of steam with similar results, and so on until a mass of water will collect, which will rush through the steam pipe and strike the first obstruction, generally the bend in the steam pipe near the cylinder, with the force of a steam hammer, and in all probabilities will carry it away and cause a disaster. This is called a **water hammer** and has caused many serious accidents.

20. Another precaution that should be taken is the easing of the throttle valve on its seat before steam is let into the main steam pipe; otherwise, the unequal expansion of the valve casing may cause the valve to stick fast and thereby give much trouble. Even if a by-pass pipe is fitted around the throttle, it would be better not to depend on it. Considerable space has been devoted to the subject of warming up and draining the water out of the steam pipe and engine on account of its importance. Water being non-compressible, it would be an easy matter to blow off a cylinder head or break a piston if the engine were started when there was a quantity of water in the cylinder.

21. The last thing for the engineer to do before taking his place at the throttle preparatory to starting the engine, provided he has no oiler, is to start the oil and grease cups feeding. It is well to feed the oil liberally at first, but not to the extent of wasting it; finer adjustment of the oiling gear can be made after the engine has been running a short time and the journals are well lubricated.

22. Cleaning Up.—After an engine has been stopped after a run and everything has been made secure, the machinery should be wiped off before the oil has had time to set. Both the bright work and the painted parts should receive attention in this respect. If there are any rusty spots on the bright work, they should be immediately scoured off with emery cloth. The floors should also be cleaned up and all dirt gathered together and consigned to the ash heap. Cleanliness is essential to a well-kept engine room, and the grade and value of the engineer in charge of it can very readily be determined by a glance of the practiced eye around the engine room. Oil should not be spilled or spattered about; there is no necessity for it and it is a waste of oil. Drip pans should be placed wherever they will do any good and they should be emptied and cleaned out at least once a day. Much saving in oil bills will be effected by the use of an efficient oil filter to filter the drip oil and use it over again instead of throwing it away.

23. When flour emery or emery cloth is used for cleaning bright work, the greatest care should be exercised not to let any of the grit get into the journals; they will be sure to cut if any substance of a gritty nature gets into them. All the oil holes in the small joints and journals that are not fitted with oil cups should be plugged up immediately after the engine is stopped and kept closed until the engine is ready to be started again. Emery should not be used to polish brass or composition; Bath brick is much better for this purpose.

STARTING A SLIDE-VALVE NON-CONDENSING ENGINE.

24. Assuming that the general instructions given in Arts. **17** to **23** have been complied with, the engineer should now take his place at the throttle, having first opened wide the stop-valves. The drain cocks on the steam pipe and engine are supposed to be open and the throttle valve just off its seat. Some steam has been allowed to enter the

valve chest and the cylinder is partly warmed up; it is now the duty of the engineer to ascertain if steam has entered both ends of the cylinder and that both ends of it are heated equally. As both steam ports cannot be open to the steam at the same time, the engine, if not provided with a by-pass, should be turned by hand so that both steam ports are opened alternately, thus admitting steam to both ends of the cylinder. In turning the engine, finally stop when the crank is on its upper or lower half center, that being the best point from which to start the engine. When it is evident that all condensation of steam has ceased in the steam pipe, valve chest, and cylinder and all the water has been blown out of them, the engine is ready to be started.

25. A slide-valve non-condensing engine is started by simply opening the throttle; this should be done quickly in order to jump the crank over the first center, after which the momentum of the flywheel will carry it over the other centers. The engine should be run slowly at first, gradually increasing the revolutions to the normal speed. When the engine has reached full speed, the drain pipes should be examined; if dry steam is blowing through them, close the drain cocks; if water is being delivered, let the drain cocks remain open until steam blows through and then close them.

26. The engineer should now make a trip around the engine to ascertain if the journals are running cool. First, try the crankpin end of the connecting-rod by touching it with the palm of the hand; to do this safely, on a high-speed engine, requires some skill and experience, but the art can be acquired by a little practice; the beginner, however, should be very cautious that he does not get his hand caught in the machinery. If the end of the connecting-rod is only blood warm, no harm has yet been done, but it is an intimation that the crankpin may get hot, and requires watching. Assuming that the crankpin is running cool, the next step is to feel the shaft journals and examine the lubricating apparatus. If the journals are running cool, decrease the

oil feed gradually and carefully until there is just enough oil fed into the journals to supply the demand without unnecessary waste. It is supposed that the engine is now running satisfactorily, and the engineer may hence turn his attention to a general inspection of his department.

27. It sometimes happens that a cracking noise is heard in the cylinder after the engine has been running for a while. This means "water in the cylinder," and the cylinder drain cocks should be opened promptly. It is also an intimation that the boiler is inclined to prime; this may be checked by closing the main stop-valve just enough to wiredraw the steam a little.

STOPPING A SLIDE-VALVE NON-CONDENSING ENGINE.

28. To stop a slide-valve non-condensing engine, it is only necessary to shut off the supply of steam by closing the throttle, but care should be taken not to let the engine stop on the center. After a little practice, the beginner will be able to stop the engine at any desired point of the revolution. No rule can be laid down for this; it is entirely a matter of experience.

29. After the engine is stopped, shut off the oil feed and close the main stop-valve; be sure that the valve is seated, but without being jammed hard down on its seat. The drain cocks on the steam pipe and engine may or may not be opened, according to circumstances. It will do no harm to allow the steam to condense inside the engine, as the engine will then cool down more gradually, which lessens the danger of cracking the cylinder casting by unequal contraction. All the water of condensation should be drained from the engine before steam is again admitted to it.

30. When an engine is required to run in either direction, in answer to signals or otherwise, as in the case of

hoisting engines and locomotives, it is usually fitted with the link-valve motion, which is operated by a system of levers or other apparatus called the **reversing gear.** In warming up an engine fitted with a link, it is only necessary to run the link up and down if a horizontal engine, or back and forth if a vertical engine, to admit steam into both ends of the cylinder; and to start or stop such an engine, either the go-ahead or the backing eccentric, as required, is thrown into action by operating the link by means of the reversing gear.

STARTING A SLIDE-VALVE CONDENSING ENGINE.

31. A steam engine of the condensing type is fitted with either a surface, a jet, or an injector condenser. The function of a condenser is to convert the exhaust steam from the engine into water by condensation, thereby producing a vacuum in the condenser. The pressure of the atmosphere is thus partially removed from the exhaust side of the piston and the net pressure correspondingly increased.

32. If the engine is fitted with a surface condenser, the condenser will be supplied with an air pump and a circulating pump. The air pump removes the air, vapor, and water of condensation from the condenser; it discharges the water into the hotwell or feed tank, while the air and vapor escape into the atmosphere. The circulating pump supplies the condensing water and forces it through the tubes of the condenser.

33. It is sometimes the case that the air pump and the circulating pump are attached to and operated by the main engine; more frequently, however, they are operated by a separate and independent steam cylinder, in which cases the apparatus as a whole, including the condenser, is called the **vacuum engine.**

Another arrangement of these pumps is the following: A centrifugal circulating pump is used instead of a reciprocating pump, by which the circulating pump can be operated independently of the air pump, permitting the speed of the circulating pump to be changed without affecting the speed of the air pump. This is desirable, because a greater quantity of injection or condensing water is required in summer than in winter on account of its higher temperature at that season of the year.

34. Before the main engine is started, the air pump and circulating pump should be put into operation and a vacuum formed in the condenser; this will materially assist the main engine in starting promptly, and in cases where the engine is worked to bell signals, such as a hoisting engine in a mine or elsewhere, this is a most important consideration. Prior to starting the air and circulating pumps, the injection valve should be opened to admit the condensing water into the circulating pump; the delivery valve should also be opened at this time. The same course of procedure that is used in warming up and draining the water out of the main engine should be followed with the vacuum engine, and it is started in the same manner, i. e., by simply opening the throttle.

35. After the main engine has been running for a few minutes to equalize temperatures, the speed of the air and circulating pumps and the admission of injection water should be regulated so as to maintain about 26 inches of vacuum in a surface condenser and a feedwater temperature of about 115° F. A higher vacuum than 26 inches, when the barometer stands at 30 inches, will result in a loss of heat from cold feedwater, and it will also cause a high-speed engine to thump while passing the centers through insufficient compression or cushion for the piston; a lower vacuum than 26 inches will cause a loss by too much back pressure. As a rule, there should be about 2 pounds (absolute) of back pressure on the exhaust side of the piston;

this is equivalent to 4 inches on the vacuum gauge and a feedwater temperature of about 115° F.; therefore, the reading of the vacuum gauge should be about 4 inches below the reading of the barometer to get the best results from the engine.

36. If an ordinary jet condenser is used, no circulating pump is required, the water being forced into the condenser by the pressure of the atmosphere. If the air pump is operated by the main engine, which is sometimes the case, a vacuum will not be formed in the condenser until after the engine is started and at least one upward stroke of the air pump is made. In this case the injection valve must be opened at the same moment the engine is started; otherwise the condenser will get "hot" and a mixture of air and steam will accumulate in it and prevent the injection water from entering. When this occurs it is necessary to pump cold water into the condenser by one of the auxiliary pumps through a pipe usually fitted for that purpose; if such a pipe has not been provided, it may be found necessary to cool the condenser by playing cold water upon it through a hose.

37. Jet condensers are sometimes fitted with a valve that automatically opens outwards, called a **snifting valve,** or **snifter,** by which the accumulated steam, vapor, and air may be discharged into the atmosphere. This valve is a disk of metal, similar to a safety valve, that is held to its seat by its own weight and the pressure of the atmosphere. It serves to relieve the condenser of pressure in case of an accident to the air pump.

38. If the engine is fitted with a jet condenser, the course of procedure in starting is similar to that followed in starting an engine with a surface condenser, viz.: The air pump should be started before the main engine is started and thereby form a vacuum in the condenser beforehand, and it should be stopped after the main engine is stopped.

STOPPING A SLIDE-VALVE CONDENSING ENGINE.

39. The operation of stopping a slide-valve surface-condensing engine is precisely similar to that of stopping a non-condensing engine of the same type, with the addition that after the main engine is stopped the vacuum engine is also stopped, and in the same way, i. e., by closing the throttle, after which the injection valve and the discharge valve should be closed and the drain cocks opened.

40. With a jet condenser, the operation of stopping the engine is the same as the above, with the exception that the injection valve should be closed at the same moment that the engine is stopped.

SUMMARY.

41. The instructions given in Arts. **24** to **40** apply to any slide-valve engine, whether vertical or horizontal, and also whether it is fitted with a ball or pendulum governor, a shaft automatic governor, or if it is without any governor at all, from the fact that a governor acts only when the engine is running at or near its normal speed; therefore, while starting or stopping an engine the governor is not in action.

STARTING A SIMPLE CORLISS ENGINE.

42. A simple engine fitted with Corliss valves and valve gear is started and stopped in a somewhat different manner from that practiced with the plain slide-valve engine.

43. In the Corliss engine the eccentric rod is so constructed and arranged that it may be hooked on or unhooked from the eccentric pin on the wristplate at the will of the engineer. The wristplate is provided with a socket for the starting bar; the starting bar may be shipped or unshipped as required.

44. In starting an engine of the Corliss type, after all the preliminaries, such as warming up and draining the water from the engine, starting the oil feed, etc., as heretofore explained, have been attended to, the starting bar is shipped into its socket in the wristplate and the throttle is opened. The starting bar is then vibrated back and forth by hand, by which the steam and exhaust valves are operated through the wristplate and valve rods; as soon as the cylinder takes steam the engine will start. After working the starting bar until the engine has made several revolutions and the flywheel has acquired sufficient momentum to carry the crank over the first center, let the hook of the eccentric rod drop upon the pin on the wristplate. As soon as the hook engages with the pin, unship the starting bar and place it into its socket in the floor.

45. The way to determine in which direction the starting bar should be first moved to start the engine ahead is to note the position of the crank, from which the direction in which the piston is to move may be learned. This will indicate which steam valve to open first; it will then be an easy matter to determine in which direction the starting bar should be moved. After a little practice the engineer will know at a glance which way to work the starting bar.

46. The engine having been started, the engineer should attend to those duties that have been mentioned in the instruction for the slide-valve engine under similar circumstances.

47. If the engine is of the condensing type, the same course of procedure in starting the vacuum engine should be followed as with the simple slide-valve condensing engine, which has been previously explained. In warming up the cylinder of a Corliss engine, it is not necessary to turn the engine to admit steam to both ends; it is only necessary to work the valves by hand with the starting bar.

STOPPING A SIMPLE CORLISS ENGINE.

48. A Corliss engine is stopped by closing the throttle and unhooking the eccentric rod from the pin on the wristplate; this is done by means of the unhooking gear provided for the purpose. As soon as the eccentric rod is unhooked from the pin, ship the starting bar into its socket in the wristplate and work the engine by hand to any point in the revolution of the crank at which it is desired to stop the engine. Then proceed as directed for the simple slide-valve engine. After stopping a Corliss condensing engine, the same course should be followed as with a slide-valve condensing engine in regard to draining cylinders, closing stop-valves, etc.

STARTING A COMPOUND ENGINE.

49. Before starting a compound engine, the high-pressure cylinder is warmed up in the same manner as a simple engine. To get the steam into the low-pressure cylinder is an operation, however, that will depend on circumstances. If the cylinders are provided with pass-over valves, it will only be necessary to open them to admit steam into the receiver and from thence into the low-pressure cylinder. If the cylinders are not fitted with pass-over valves, the steam can usually be worked into the receiver and low-pressure cylinder by operating the high-pressure valves by hand. Sometimes compound engines are fitted with starting valves, which greatly facilitate the operations of warming up and starting. Usually a compound engine will start upon opening the throttle.

50. It sometimes happens that the engine will refuse to start from various causes, viz.: The high-pressure crank may be on either dead center; there may be too low a steam pressure in the receiver; the engine may be stiff from standing idle for a long time, and the oil in the journals has become gummy; the pistons may be rusted fast in the cylinders, or the cylinders may not have been wiped out

after the last run and a coating of carbonized oil and rust may have collected on their walls and caused the pistons to stick fast. If the engine is fitted with independent adjustable cut-offs, the cut-offs may be set to cut off too early; or there may be water in the cylinders. There may be some obstruction to the engine turning, although that matter is supposed to have been attended to during the preliminary inspection. In most cases the conditions will suggest the remedy. If the high-pressure crank of a cross-compound engine is on its center and the low-pressure engine will not pull it off, it must be jacked off. Ordinarily, it will be found that upon admitting steam of sufficiently high pressure into the receiver, the low-pressure piston will move and take the high-pressure crank from off the center; if the pressure in the receiver is too low to start the low-pressure piston, more steam must be admitted into the receiver. If the engine is stuck fast from gummy oil or rusty cylinders, all wearing surfaces must be well oiled and the engine jacked over at least one entire revolution. If the cut-offs are run up, run them down, full open. If there is water in the cylinders, blow it out through the cylinder relief or drain valves, and if there is any obstruction to the engine turning, remove it.

51. If the crank of a tandem compound engine is on the center, it must be pulled or jacked off. If the high-pressure crank of a cross-compound engine is on the center, it may or may not be possible to start the engine by the aid of the low-pressure cylinder, depending on the valve gear and crank arrangement. When the cranks are 180° apart (which is a very rare arrangement), the crank must be pulled or jacked off the center. When the cranks are 90° apart and a pass-over valve is fitted, live steam may be admitted into the receiver and thence into the low-pressure cylinder, in order to start the engine. When no pass-over is fitted, but the engine has a link motion, sufficient steam to pull the high-pressure crank off the center can generally be worked into the low-pressure cylinder by working the

links back and forth. When no pass-over is fitted, but the high-pressure engine can have its valve or valves worked by hand, steam can be gotten into the low-pressure engine by working the high-pressure valve or valves back and forth by hand. If no way exists of getting steam into the low-pressure cylinder while the high-pressure crank is on a dead center, it must be pulled or jacked off.

52. If the air and circulating pumps are attached to and operated by the main engine, a vacuum cannot be generated in the condenser until after the main engine has been started. Consequently, in this case, there is no vacuum to help start the engine; therefore, if it is tardy or refuses to start, it will be necessary to resort to the jacking gear and jack the engine into a position from which it will start. With an independent vacuum engine, however, it is seldom that any such difficulties in starting an engine are encountered. A vacuum having been generated in the condenser beforehand, the pressure in the receiver acting on the low-pressure piston causes the engine to start very promptly, even though the high-pressure crank may be on its center. Notwithstanding differences of opinions among designers in regard to this matter, the value of an independent vacuum engine is fully appreciated by the man at the throttle. In fact, it is almost indispensable with compound condensing engines of high power that are required to run in either direction in answer to bell or other signals, such as hoisting, rolling-mill, and marine engines, where promptness in starting is absolutely essential.

53. Large reversible engines are usually fitted with steam starting and reversing gears, each builder suiting his own fancy in regard to the design; therefore, the engineer should promptly familiarize himself with the mechanism of the particular starting gear that he has to handle.

54. If the engine is fitted with adjustable cut-offs, they should be so manipulated, after the engine has been started,

that it will run smoothly and at the lowest steam consumption attainable with the given load. The only way of finding the proper points of cut-off is by experiment, setting the cut-offs where judgment and experience dictate and noting the effect upon the smooth running and steam consumption.

STOPPING AND REVERSING A COMPOUND SLIDE-VALVE ENGINE.

55. Compound slide-valve engines, whether condensing or non-condensing, are stopped by closing the throttle, and, if a reversing engine, throwing the valve gear into mid-position. If the stop is a permanent one, the usual practice of draining the engine, steam chests, and receiver, closing stop-valves, stopping the oil feed, etc. may be followed; or, as before stated, if the cylinders and receivers are complicated castings, as they are apt to be in an engine of this kind, it would be better not to drain them while they are hot, but to let them cool down gradually to avoid the danger of cracking the castings from too sudden and, therefore, unequal contraction.

56. If the engine is intended to run in both directions in answer to signals, as in the cases of hoisting, rolling-mill, and marine engines, the operator, after stopping the engine to signal, should immediately open the throttle very slightly, in order to keep the engine warm, and stand by for the next signal. If the engine is fitted with an independent or adjustable cut-off gear, it should be thrown off, i. e., set for the greatest cut-off, for the reason that the engine may have stopped in a position in which the cut-off valves in their early cut-off positions would permit little or no steam to enter the cylinders, in which case the engine will not start promptly, and perhaps not at all. While waiting for the signal, the cylinder drain valves should be opened and any water that may be in the cylinders blown out. When dry steam blows through the drains, the cylinders are clear of water.

57. When the signal to start the engine is received, it is only necessary to throw the valve gear into the go-ahead or backing position, as the signal requires, and to operate the throttle according to the necessities of the case, for which no rule can be laid down beforehand, as the position of the throttle will depend on the load on the engine at the time. Handling the throttle must be learned by experience on the spot.

58. It is frequently the case that in large plants a working platform is provided on which the reversing gear, throttle-valve lever or wheel, cylinder drain-valve levers, and all other hand gear, gauges, etc., are located and placed within easy reach of the engineer's station. This platform is usually placed in a commanding position, from whence the engineer has a full view of the moving parts of the engine. This is a matter of considerable importance, although an experienced engineer, after he becomes familiar with the various sounds produced by his engine under different conditions, will depend on the ear as much as on the eye in running it. In most cases any derangement of the machinery will give timely warning by making an unusual sound; perhaps it may be only a slight clicking noise, scarcely noticeable among so many different sounds. A careful engineer, however, can detect it as quickly as an expert musician can detect a discordant note, and he should at once proceed to find out the cause, thereby anticipating and preventing a possible breakdown.

STARTING AND STOPPING A CORLISS COMPOUND ENGINE.

59. The operation of starting and stopping a Corliss compound engine is precisely similar to that of starting and stopping a Corliss simple engine; the high-pressure valve gear only is worked by hand in starting, the low-pressure eccentric hook having been hooked on previously. The low-pressure valve gear is only worked by hand while

warming up the low-pressure cylinder. The same directions that were given for operating the simple condensing engine apply to the Corliss condensing engine, so far as the treatment of the air pump, circulating pump, and condenser is concerned.

STARTING, STOPPING, AND REVERSING TRIPLE- AND QUADRUPLE-EXPANSION ENGINES.

60. The management of triple- and quadruple-expansion engines is the same as that practiced with the compound engine, with the exception that there is a greater number of moving parts, more journals, more hand gear, and more machinery, in general, to look after, requiring greater activity and alertness on the part of the engineer to care for it.

LINING ENGINES.

INTRODUCTION.

61. Purpose.—The operation of lining an engine is for the purpose of locating the different parts in relation to each other, so that no undue strains will be exerted on any one of the parts and the friction of the moving parts will be reduced to a minimum. Absolute accuracy, though desirable, cannot be attained, and if it could be attained, it could not be maintained. Too much care, however, cannot be exercised in lining an engine, as its future smooth running and efficiency will depend very largely on the accuracy with which this operation is performed.

62. Requirements.—An engine, in order to be in line, must fulfil the following requirements:

1. The center line of the shaft must be at right angles to the center line of the cylinder.

2. The wearing surfaces of the guides must be parallel to the center line of the cylinder. When two guides are used, they must be parallel to each other, and, at least in most designs, equidistant from the center line of the cylinder.

3. The center line of the wristpin must be at right angles to the center line of the cylinder and must lie in the same plane.

4. The center line of the crankpin must be parallel to the center line of the shaft.

5. The center line of the cylinder and of the shaft must both lie in the same plane.

6. The center line of the bore of the brasses at both ends of the connecting-rod must be parallel to one another and must be at right angles to the center line of the connecting-rod.

7. The center line of the piston rod must coincide with the center line of the cylinder.

If the above requirements are fulfilled, the engine may be said to be in line, as far as the machine itself is concerned. In addition to the requirements enumerated above, it is generally necessary that the crank-shaft be level.

LINING UP.

63. Preliminary Conditions.—Let it be understood that a new single-cylinder, horizontal engine is in the course of erection or that an old engine of the same type is receiving a thorough overhauling. In the case of the new engine, it is assumed that the foundation has been built, the bedplate placed in its proper position and secured there by the anchor bolts, and the cylinder has been secured to the bedplate. As the cylinder was fitted to the bedplate in the shop, it may be assumed that it is correctly placed. In the case of an old engine being overhauled, it is understood that all the moving parts have been removed and that the cylinder heads are off. The condition of both engines are now supposed to

be the same; therefore, henceforth the course of procedure will be the same for both.

64. Stretching Center Line of Cylinder.—The first step in lining an engine is stretching a line coincident with the center line of the cylinder. This may be done in the following manner:

A strip of board or other convenient material is secured to the head end of the cylinder by means of the stud bolts and nuts, as shown at *a*, Fig. 1. A hole about 1 inch in diameter

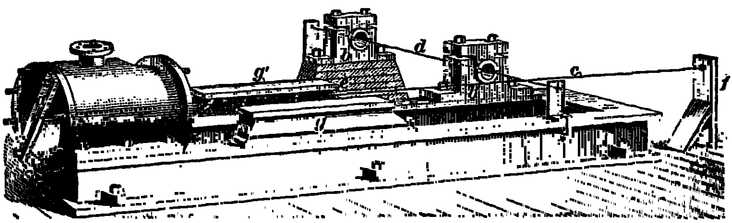

FIG. 1.

is bored through the strip approximately in line with the center of the cylinder. Some form of standard, as *f* for instance, which is pierced similarly to the strip of board *a*, should be erected at the crank end of the bedplate. A very fine braided cord or piece of thin annealed copper wire, as *c*, may now be stretched very tightly through the holes in *a* and *f*. In order to allow of ready adjustment, each end of the line may be fastened to the middle of a piece of, say, $\frac{1}{4}$-inch round iron about 2 inches long, or, better yet, be passed through a hole in the center of a piece of stout sheet tin or other metal. A knot at each end of the line will prevent it slipping through the holes. The pieces of sheet metal may be cut, say, 2 inches square, and the diagonally opposite corners may be turned up at right angles to form handles by which they may be adjusted. If the line used is a fine wire, two holes may be punched in each piece of sheet metal and the end of the wire passed through one of the holes, brought back through the other in the form

of a loop, and the end stopped off around the main part of the line.

65. The line should be set central to the bore of the cylinder at the head end by calipering from the inside of the cylinder counterbore to the line. This may be done with a pair of inside calipers, but in most cases it is better to use a light pine stick, like that shown in Fig. 2. The stick *a* should be about 1 inch shorter than the radius of the cylinder counterbore and tapered at each end, with a thin 1-inch

FIG. 2.

wire nail driven into each end, as shown at *b*, making the total length of the stick, including the nails, the exact radius of the counterbore. The advantage of the stick in calipering is that it is lighter and more convenient to use than inside calipers.

If the calipers or stick will just touch the line, no matter from which point on the circumference of the counterbore the measurement is taken, the head-end part of the line will be central to the bore of the cylinder, provided the measurements were carefully and accurately made. If the measurements do not agree, the line that passes through *a*, Fig. 1, must be shifted in the direction shown by the variation until it coincides with the center line of the cylinder. It is considered good practice to make four measurements 90° apart.

66. After adjusting the line at the head end of the cylinder, the crank-end part of the line may be trued up in a similar manner by moving the line at the standard *f*, Fig. 1. The line, now, may or may not be properly adjusted. Hence, to make sure, the alinement of the line at the head end should be tried again. Now, unless the line happened to be very close to the center line of the cylinder before any adjustments were made, it will usually be found to be a shade out of the center, and hence requires readjustment. After adjusting the head-end part of the line, try

the crank-end part again and adjust it. Continue this practice, first at one end of the cylinder and then at the other, until no further adjustment is necessary or possible. Then, if the measurements have been carefully made, the line *c* may be considered to coincide with the center line of the cylinder.

67. If the crank-end head is cast solid with the cylinder, as is frequently the case, the line *c* must be trued up from the bore of the piston-rod stuffingbox. This may be done by means of a stick similar to that shown in Fig. 2, but it may be more conveniently done by means of the device shown in Fig. 3. This device consists of a hardwood block *a*, which is turned to just fit into the stuffingbox and has a ¼-inch hole *b* bored in the exact center. The face of the block is faced square with the outside, and two center lines *c d* and *e f* are drawn across the face at right angles to each other. By sighting along these lines, it is easy to determine when the line or wire is central with the stuffingbox.

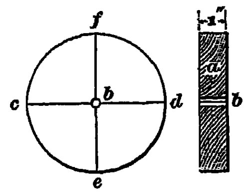

FIG. 3.

68. Stretching Center Line of Shaft.—After completing the adjustment of the line *c*, Fig. 1, another line, as *d*, is stretched to coincide with and to represent the center line of the shaft; it *must* be at right angles to the line *c* and generally *must* be level.

In many engines, the outboard bearing *b*, Fig. 1, is movable to a certain extent and may be adjusted in regard to its relative position with the inboard bearing *b'*. The inboard bearing, however, is usually cast solid with the bedplate; therefore, it is fixed and cannot be moved. Hence, in adjusting the line *d*, the aim must be to stretch the line through the center of the bearing *b* and at the same time to have it at right angles to the line *c*. In order to accomplish this, it will be necessary to erect two standards, as shown in Fig. 1, to which to fasten the line. It is supposed that the top brasses and caps of the main-shaft bearings are in place

just as they would be if the shaft itself were in place. The alinement of the line *d* in reference to the bearing *b'* can be tested by calipers, but a better method is to insert a block of wood, made as shown in Fig. 3, into the bore of the bearing. It would be well to fit a similar piece of wood into the bore of the outboard bearing also, as it will greatly facilitate the adjusting of the line *d* by passing it through the holes in the centers of these blocks.

69. Squaring Center Line of Shaft.—We may now proceed to test the angle between the two lines *c* and *d*, Fig. 1. The line *d* may be approximately squared with the line *c* by a carpenter's square pressed gently against the two lines. Great care in the use of the square is necessary, since the lines will yield to quite an appreciable extent under a very slight pressure. A crowding of some part of the square against either line will deflect it and seriously interfere with the test. If the square shows that the lines are not at right angles to each other, the line *d* should be shifted until they are, always keeping in mind the fact that the line must coincide with the center line of the inboard bearing.

70. In lining engines of the larger sizes, the carpenter's square is not accurate enough, since its legs are very short in proportion to the length of the lines. A somewhat different method may then be used, which is based upon the principles of geometry.

Procure a slender strip of wood—6, 8, or 10 feet long, according to the extent of the space to work in. Taper the strip to a point, like a lead pencil, at each end; divide the strip into two exactly equal parts and mark it plainly in the middle. Now hold the strip gently along the line *c*, Fig. 1, with the mark in the middle at the line *d*; mark the line *c* at each end of the strip; then put one end of the strip at one of the marks just made on line *c* and the other end of the strip against the line *d* and mark the place where the end of the strip touches the line *d*; repeat this

operation on the other side of the line *c*, and if the end of the strip touches the line *d* at exactly the same spot that it did before, the line *d* is at right angles to the line *c*; but if the end of the strip does not touch the same spot on the line *d* at both trials, the line *d* is not at right angles to the line *c*, and the line *d* must be shifted in the right direction and the whole operation must be repeated until the end of the strip *does* touch the line *d* at the same point at both trials. In shifting the line *d* during these operations, care must be taken that its position through the center line of the inboard bearing *b'* is maintained. A convenient method of marking the lines is to tie a piece of bright-colored thread around the lines at the points that are desired to be marked. The colored threads are easily sighted, and the marks are more sharply defined and cleaner than when made with paint or chalk and they have the advantage of being easily shifted along the lines at will.

71. In order to prevent any displacement of the line while measuring, it is good practice to place blocks of wood or any other convenient material against it at the points marked, in order to steady it, but care must be used not to deflect the line; when the blocks are properly placed, they may be temporarily secured in position.

72. Another test to determine whether the lines are at right angles to each other is to measure from their intersection distances of 6 and 8 feet, one on each line. Then, if the measurement from line to line, measuring in a straight line from the points just laid off, is 10 feet exactly, the lines are at right angles. Instead of using the values 6, 8, and 10 feet, any convenient multiple may be used.

73. Leveling Center Line of Shaft.—The line *d* representing the center line of the crank-shaft may be tested for being level by means of a spirit level, taking great care in applying it not to deflect the line. Another method is to drop a plumb-line from overhead touching the line *d*.

Then, if the line *d* is level, it evidently must be at right angles to the plumb-line, and, consequently, this condition can be tested in the same manner in which the position of line *d* in reference to line *c* was tested.

74. In leveling the line *d*, Fig. 1, it should not be allowed to touch the line *c*, lest it should be deflected; it should pass just over or just under it, say at a distance of $\frac{1}{100}$ inch. If the lines touch each other, any vertical movement of either end of the line *d* will, when in the wrong direction, deflect both lines, thus defeating the primary object of stretching them; viz., that they shall represent the center lines of the cylinder and shaft. After leveling the line *d*, it is well to verify the relative alinement of both lines. When the line *d* is properly adjusted, the fifth requirement will be complied with a degree of accuracy sufficient for practical purposes, providing the lines *d* and *c* will just clear each other where they cross.

75. In lining up a new engine or an old engine with new shaft-bearing brasses, it is generally desirable to have the center line of the shaft lay about $\frac{1}{32}$ inch above the center line of the cylinder, so that as soon as the brasses and journals have worn down to their bearings, the center line of the shaft will be very nearly level with the center line of the cylinder.

76. Shifting Outboard Bearing.—Having proved the correct alinement of the lines *c* and *d*, the outboard bearing may now be shifted until the line *d* coincides with its center line, when it may be secured in position permanently.

77. Testing Alinement of Guides.—We are now ready to test the guides *g*, *g'*, Fig. 1. If the line *d* is level, a spirit level may be used to ascertain if they are in the proper plane in reference to *d*. Their adjustment relative to the center line of the cylinder may be tested in one direction by placing a straightedge successively at each end of the

guides and squarely across them. Then, if the measurements from the lower edge of the straightedge down to the line *c* agree at both ends of the guides, they are in line vertically. If not, the same ends of both guides must either be raised or lowered, remembering that raising one end of the guide is equivalent to lowering the other end. By measuring from the inside edges of the guides to the line *c*, it may be ascertained if the guides are parallel to each other and parallel to the center line of the cylinder. To ascertain if the guides are in a horizontal plane, a spirit level may be placed squarely across the guides at both ends.

78. In lining the guides of a Corliss engine a special device similar to that illustrated in Fig. 4 is often used. This consists of a casting *a* that is turned to fit the inside of

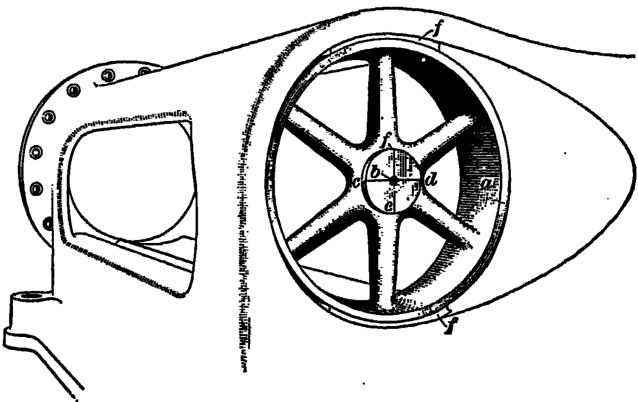

FIG. 4.

the guides. At the center there is a small hole *b* through which the line passes, while the lines *c d* and *e f*, drawn at right angles to each other, serve to locate the center line in its proper position, this being done in a manner similar to that illustrated in Fig. 3 for the piston-rod stuffingbox.

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79. Bedding the Shaft.—The line *d*, Fig. 1, having been removed, the crank-shaft, crank, and flywheel are put in place, care being taken not to disturb the line *c*. After the bearings have been adjusted so that the shaft will turn easily in them, the journals of the shaft should be wiped clean and given a coat of red or black marking material. The shaft should then be placed in its bearings, with the lower halves of the brasses in position, and rocked back and forth a few times. The shaft is then lifted out of the bearings and the high spots scraped off with a half-round scraper. This operation is repeated until the shaft shows a good bearing in both the main pillow-block and the outboard bearing. After the lower halves of the boxes are scraped, the upper halves may be put in place and fitted in like manner. The shaft is now lifted from its bearings and the eccentrics and governor-driving device are placed in position, after which the shaft is returned to its place. The crank and flywheel having been fitted to the shaft in the shop, it is to be presumed that they are true with the shaft.

80. Testing Alinement of Shaft.—In order to make sure that the shaft is exactly at right angles to the center line of the cylinder and that it is also level, the following course

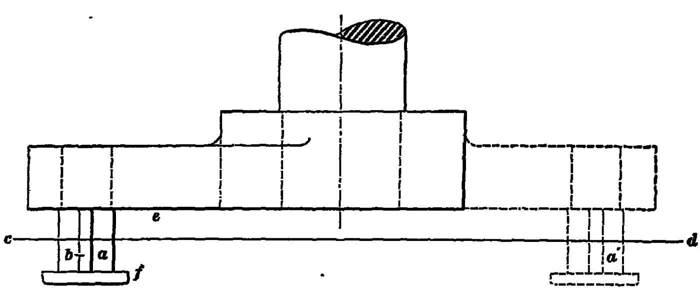


FIG. 5.

may be pursued: The crankpin *a*, Fig. 5, is brought up to the center line *c d* and a piece of wood *b* is fitted between

the face of the crank *e* and the head of the crankpin *f*. A mark is made on this piece of wood to coincide with the line *c d*. The shaft is now given a half revolution to bring the crankpin under the line at the other end of its travel, as shown by the dotted lines at *a'*. If the line on the strip of wood *b* again coincides with the center line *c d*, the shaft is at right angles to the center line of the cylinder.

81. Testing Leveling of Shaft.—In order to test the shaft to see whether or not it is level, a fine plumb-line may be suspended vertically before the shaft at the crank end and the crankpin *a* brought into contact with it at the upper half-center and then tested again at the lower half center. If the end of the crankpin just touches the line at both upper and lower half centers, the shaft is horizontal.

82. Lining the Crosshead.—The line *c*, Fig. 1, having been removed, the piston, piston rod, and crosshead are put in place, centering the piston in the cylinder first of all, when the design of the piston is such as to make this adjustment necessary. The next step to take is to ascertain if the center line of the piston coincides with the center line of the cylinder. This may be done as follows: In Fig. 6, let *d* be the upper surface of the guides, which, when properly alined, lies parallel to the center line of the cylinder; the piston having been previously centered, the center line of the piston rod coincides with the center line of the cylinder; therefore, the piston rod should be parallel with the upper surfaces of the guides. This may be readily tested by placing the piston at the forward end of its stroke; then measure downwards from the lower edge of a straightedge laid across the guides at *e* and *f* to the piston rod. If the

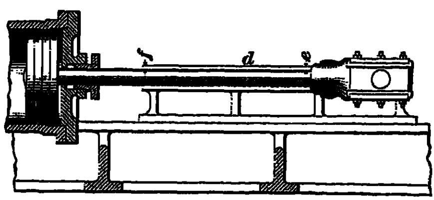


FIG. 6.

two measurements agree, the piston rod is in line; otherwise, the crosshead shoes must be adjusted until the piston rod is in its proper position.

83. Testing the Crankpin.—A method of testing the accuracy of the crankpin is shown in Fig. 7 and it may be put into practice as follows: Connect the connecting-rod to the crankpin and key up the brasses snugly to the pin; then put the crank on or near one of its dead centers, as at c, Fig. 7, and exert a slight pressure against the wristpin end of the connecting-rod in the direction of the arrow x, in order to take up any lost motion in the crankpin brasses. It is understood that the connecting-rod is disconnected from the crosshead, and the latter is pushed back towards the cylinder, so as to be out of the way; then measure the distance a and make a note of it. Put the crank on or near the other dead center c' and measure the distance a'. Then, if the distance a is equal to the distance a', it proves that the center line of the crankpin is parallel to the center line of the shaft in the horizontal plane. But it is necessary that similar conditions should prevail in the vertical plane. To test this, put the crank on the upper half center, as at d, and measure from the wristpin end of the connecting-rod to the inner edge of the guides, as before; then turn the crank to the lower half center d' and measure again from the connecting-rod to the guides. If the measurements agree for both positions of the crank, the crankpin is properly alined. In the figure the crankpin is shown out of line, the connecting-rod then occupying the positions shown in full

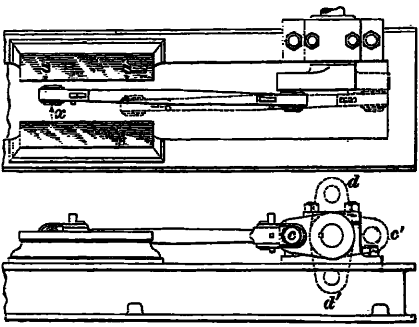


FIG. 7.

and dotted lines, respectively. If the error is very small. it may sometimes be remedied by filing and scraping the crankpin, but if the error is serious, it may require machine work.

84. Testing Bore of Crankpin and Wristpin Brasses. It will generally be advisable to ascertain if the bore of the crankpin brasses is at right angles to the center line of the connecting-rod. This may be done by putting the crank, with the connecting-rod attached to the crank but disconnected from the crosshead, on one of its dead centers, as at *c*, Fig. 7. Then measure the distance from the inner edge of one of the guides to the crosshead end of the connecting-rod, as at *a*, Fig. 7, and make a note of it; then take the rod off the pin and turn the rod half way around on its center line and replace it on the pin. Now measure from *a*, as before; if the two measurements agree, the brasses are correctly bored. To make this test still more satisfactory, the operation may be repeated for several different positions of the crank. The wristpin end of the rod may be tested in like manner; in this case the rod is disconnected from the crankpin and the measurements are taken from the face of the crank or from the collar of the crankpin to the crankpin end of the connecting-rod. If the connecting-rod cannot pass this test satisfactorily, the brasses must be fitted to the crankpin and wristpin by chipping, filing, and scraping until it fills the requirements.

85. Testing Alinement of Wristpin.—The alinement of the wristpin may now be tested, for which purpose the connecting-rod may be used. Key the rod rather snugly to the wristpin, having it disconnected from the crankpin. Then place the crank on or near one of its dead centers and push the crosshead forwards until the end of the connecting-rod just rests on the crankpin. If the center line of the rod is the same distance from both collars of the crankpin, the wristpin is in the correct position in one direction. To

prove it for another direction, put the crank on one of its half centers and repeat the above described operation. If the result is the same as before, the wristpin is in its correct position relative to the center lines of the cylinder and shaft.

86. Testing Alinement of Connecting-Rod Brasses in Reference to Each Other.—It still remains to be proved that the center lines of the crankpin brasses and the wristpin brasses lie in the same plane. This is a very necessary condition, because, otherwise, the brasses when keyed tightly to one pin will not fit the other pin, or, as usually expressed, they will bear on one side only. To make this test, a thin coating of Prussian blue or red-lead paint is put on the crankpin and the rod is connected up and adjusted rather snugly to the wristpin and a little less snugly to the crankpin. The crank is then turned through one revolution, when the crankpin brasses may be examined. If they show marking all over, their correct adjustment is assured, but if the coloring matter is rubbed off at either end of the crankpin, it shows that the brasses do not fit the pin and that they must be filed and scraped until they bear equally on all parts of the pin.

87. Order of Operations.—An engineer in lining up his engine or testing his engine for alinement, will do well to perform the various operations in the same order as given here; he should remember that in order to insure correct results, each part of the engine tested *must* be alined before proceeding further. Thus, it is folly to attempt to prove by the methods given here that the center line of the brasses are in the same plane before the correct relative alinement of the wristpin and crankpin are proven.

If a new engine is so far out of line that it cannot readily be adjusted by liners and scraping brasses while the various parts are being assembled, it should be placed in good order by the builder.

POUNDING OF ENGINES.

CAUSES.

88. The causes of pounding in engines are various; they are not always easy to locate in a large engine, owing to the difficulty of locating the exact source of the sounds. These sounds serve as a warning, however, that something is wrong about the machinery, and no time should be lost in ascertaining where and what it is and taking measures to stop it, thereby preventing a possible breakdown.

LOOSE BRASSES.

89. The most frequent cause of pounding in engines is loose journal brasses; the pounding is produced by the journals striking against the sides of the brasses as the cranks are passing the centers and at the instant the change of direction in the motion of the pistons takes place. If the journal-boxes are very slack, the pounding may be so violent as to cause heating of the journal and boxes by the succession of blows they receive; this may especially occur at the crankpins. The remedy for pounding of this nature is obvious. Stop the engine and set up on the brasses gradually, until, after trial, the pounding ceases, taking great care that they are not set up too tight, else they will heat from friction, which may have a more disastrous effect than a moderate amount of pounding. In the case of shaft journals, they may be set up without stopping the engine, provided they can be reached without danger of the engineer being caught in the machinery.

90. It may so happen that the boxes or brasses are worn down until the edges of the upper half and the edges of the lower half are in contact and cannot be set up on the journal any farther; they are then said to be **brass and**

brass, or brass bound. In a case of this kind, the journal must be **stripped,** as it is called, when the cap and brasses are removed from a journal. The edges of the brasses are then chipped or filed off, in order to allow them to be closed in; the amount to be taken off may be determined by trying the brasses on the journal occasionally or by calipering the journal with outside calipers, transferring the measurement to a pair of inside calipers, with which to measure the bore of the brasses as they are being fitted. It is a most excellent plan in practice to reduce the two halves of the brasses so that they will stand off from each other when in place for a distance of $\frac{1}{8}$ inch to $\frac{3}{16}$ inch and fill this space with hard sheet-brass liners, say from 20 to 22 Birmingham wire gauge in thickness each. The object is this: Should the journal become brass bound, the cap may be slacked off and a pair of the liners slipped out without the necessity of stripping the journal, which it is desirable to avoid whenever possible for the reason that it seems to be impossible in practice to put the journal brasses back just where they were before they were disturbed. In large engines it is almost always the case that journals *will* heat after being stripped, and they require a special watch for several days or until they settle down to their proper position relative to the journal.

91. In some instances journal-boxes are fitted with **keepers,** or **chipping pieces,** as they are sometimes called. These consist usually of a cast-brass liner, anywhere from $\frac{1}{4}$ inch to $\frac{1}{2}$ inch in thickness, having ribs or ridges cast on one side, for convenience of chipping and filing. These keepers are sometimes made of hardwood and are capable of being compressed slightly by the pressure exerted upon them during the setting-up process. When the boxes are babbitted, the body of the box is occasionally made of cast iron, in which case iron liners and keepers are used instead of brass ones.

92. The crankpins, being the journals most liable to heat either from pounding or from friction caused by the brasses

being set up too tightly, and on account of the comparatively small surface over which the friction is distributed, require the greatest care and need constant watching. The oiling device should be of the best and the oil should never be permitted to stop feeding or the oiling device to get out of order, else there will be trouble.

LOOSE THRUST BEARING.

93. In engines fitted with some types of friction couplings, there is a thrust exerted upon the shaft in the direction of its length. This will necessitate having a **thrust bearing,** or **thrust block,** as it is sometimes called. There is a variety of thrust bearings, but the most common is the collar thrust, which consists of a series of collars on the shaft that fit in corresponding depressions in the bearing. If these collars do not fit in the depressions rather snugly, the shaft will have end play and there probably will be more or less pounding or backlash at every change of load on the engine. This can only be remedied by putting in a new thrust bearing and making a better fit with the shaft collars, unless the rings in the bearing are adjustable, as is sometimes the case, when, of course, the end play may be taken up by adjusting the rings.

WATER IN CYLINDER.

94. Pounding often occurs in the cylinders and is frequently caused by water, due to condensation or carried over from the boilers. This may be a warning that priming is likely to occur in the boilers or has already commenced. The first thing to do at such a time, if the cylinders are not fitted with automatic relief valves, is to open the drain cocks as quickly as possible and to close down the throttle a little to check the priming.

If boilers show a chronic tendency to prime, it is because they are too small for the engine, or they have not steam space enough, or the water may be carried too high

in them, which will cause a considerable reduction of the steam space. Unsteady firing, producing great fluctuations in the steam pressure, will also cause both foaming and priming, the result of which is that water will be carried over from the boilers into the cylinders. This is always a source of danger.

Water is non-compressible; therefore, after the clearance space of the cylinder is filled and more water is allowed to enter, if there is no way for it to escape, either the cylinder head will be blown out or the piston broken. Partly closing the stop-valve of the boiler showing a tendency to prime, thereby wiredrawing the steam a little, will generally check priming, if the remedy is applied before the priming becomes violent, after which it is difficult to suppress.

LOOSE PISTON.

95. Another source of pounding in the cylinder is that the piston may be loose on the rod; this is caused by the piston-rod nut or key backing off or the riveting becoming loose, permitting the piston to play back and forth on the piston rod. If due to the nut backing off, the engine should be shut down instantly on its discovery. There is very little room to spare generally between the piston-rod nut and the cylinder head; therefore, it cannot back off very far before it will strike and break the cylinder head. After the engine is stopped and the main stop-valve closed, take off the cylinder head and set up on the piston nut as tightly as possible; there is usually a socket wrench furnished with each engine expressly for this purpose.

96. Although piston-rod nuts seldom work loose and those of vertical engines are less liable to do so than others, still as a measure of safety a taper split pin should in all cases be fitted through the piston rod behind the nut or a setscrew fitted through the nut. If, on examination, this setscrew is found slack, the cause of the nut backing off is thereby explained, and it should be screwed down solid to prevent a recurrence of the trouble.

SLACK FOLLOWER PLATE.

97. A slack **piston follower plate,** or **junk ring,** as it is called by English engineers, will cause pounding in the cylinder. It seldom happens, however, that *all* the follower bolts back out at one time unless they fit very loosely in their sockets, but it is not an infrequent occurrence that one of the follower bolts works itself out altogether and swashes about the cylinder at random. This is a very dangerous condition of affairs, especially in a horizontal engine. If the bolt should get "end on" between the piston and cylinder head, which it surely will sooner or later, either the piston or the cylinder head is bound to be broken. Therefore, if there is any intimation that a follower bolt is adrift in the cylinder, shut down the engine instantly, take off the cylinder head, remove the old bolt, and put in one having a tighter fit.

BROKEN PISTON PACKING.

98. Broken packing rings and broken piston springs will cause a great noise in the cylinder, but it is more of a rattling than a pounding noise, and the sound will easily be recognized by the practiced ear. There is not so much danger of a serious breakdown from these causes as may be supposed, from the fact that the broken pieces are confined within the space between the follower plate and the piston flange. Although they rattle around in the cylinder and make a startling din, they cannot get out or do much harm, aside from causing a leaky piston in the case of the packing rings breaking or possibly slightly scoring the cylinder face. As a matter of course, this should be repaired as soon as possible.

PISTON STRIKING HEADS.

99. There is another source of pounding in the cylinder that is usually confined to old engines; it is produced by the piston striking one or the other cylinder head. One of

the causes of this is the wearing away of the connecting-rod brasses. Keying up the brasses from time to time has the effect of lengthening or shortening the connecting-rod, depending on the design, and this change in length destroys the clearance at one end of the cylinder by an equal amount. The remedy is to restore the rod to its original length by placing sheet-metal liners behind the brasses; this obviously will move the piston back or ahead and restore the clearance. It may be mentioned here that the length of a connecting-rod is measured from center to center of the bore of the crankpin and wristpin brasses.

It is obvious that the piston should never be allowed to strike the cylinder head. This condition generally is not reached suddenly; it is brought about gradually, covering a considerable period of time amply sufficient to forestall any such occurrence. A rather rare case of the piston striking the cylinder head is due to the piston rod unscrewing from the crosshead, in case it is fastened by a thread and check-nut. To obviate any danger, the check-nut should be tried frequently.

Every reciprocating steam engine should have the length of the stroke and the clearance space at each end of the cylinder marked on the guides; by this means the relative positions of the piston and cylinder heads can always be seen at a glance.

IMPROPER STEAM DISTRIBUTION.

100. The primary cause of another source of pounding is the improper setting of the steam valve, or possibly its improper design, as it cannot always be accepted as granted that the valve is properly designed. In the case of improper setting of the valve, insufficient compression, insufficient lead, cut-off too early, and late release may all cause pounding on the centers.

101. The manner in which insufficient compression causes pounding may be explained as follows: For practical

reasons there must always be some lost motion at the wrist-pin, crankpin, and shaft bearings. Now, in passing the dead centers, the direction of pressure is suddenly reversed, and in consequence the piston rod, connecting-rod, and crank-shaft will be suddenly thrown forwards by the inrushing steam to an extent depending on the lost motion at the pins and shaft bearing. It is this sudden changing of the lost motion from one brass to another, with a violence that may be likened to a blow, that causes an engine to knock in passing the centers when compression is insufficient.

102. The effect of a reversal of pressure is clearly shown in Fig. 8. With the crankpin at *a* and the engine running

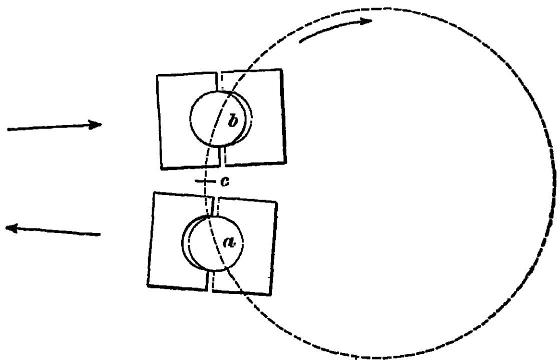


FIG. 8.

over, the connecting-rod is subjected to a pull, but after the crankpin has passed the dead center *c*, the connecting-rod is subjected to a push, in which case the rear brass, as shown at *b*, bears against the crankpin, while in the former case, as shown at *a*, the front brass bears against the crankpin.

By giving a sufficient amount of compression, the lost motion in the pins and journals is transferred gently from one side to the other before the crankpin reaches the dead center, so that by the time the live steam suddenly acts on the piston it cannot throw the rod forwards. If the compression is insufficient to gently take up the lost motion, there will be pounding.

103. Too much compression causes such a great resistance to the motion of the crank that it will tend to slow it down and thus increase the unsteadiness of the engine. Abnormal compression manifests itself by a dull, muffled sound in the cylinder or on an indicator card by the compression line rising above the steam line. It may cause pounding at the journals.

104. Insufficient lead is a common cause of pounding; in fact, it is rare to see an indicator card that shows sufficient steam lead. The exact amount of lead to be given to prevent pounding can only be determined by an actual trial; in general, slow-speed engines will require less lead than high-speed engines. In most engines the lead can be readily changed by a proper adjustment of the valve gear. In automatic cut-off, high-speed engines of the shaft-governor type, however, it is not possible, as a general rule, to change the lead by any simple adjustment, the lead having been fixed by the builder, and a change of it will require an extensive rebuilding of the governor.

105. The reason that insufficient lead causes an engine to pound is because the piston has then little or no cushion to impinge upon as it approaches the end of its stroke, and it is brought to rest with a jerk, as it were. A similar effect will be produced by a late release; the pressure is retained too long on the driving side of the piston. The ideal condition is that the pressures shall be equal on both sides of the piston at a point in its travel just in advance of the opening of the steam port. The position of this point varies with the speed of the piston and other conditions that the indicator card only can reveal; in fact, all conditions dependent on the set of the steam valve can be investigated only by the help of the indicator card. Any departure from the ideal condition above mentioned will produce more or less pounding in an engine.

106. A too early cut-off will expand the steam down too low—even below the back-pressure line sometimes.

This is an abnormal condition, which will cause pounding, and should not be permitted to occur.

A very high vacuum in a condensing engine will sometimes cause pounding by not permitting sufficient cushion for the piston to impinge upon.

POUNDING AT CROSSHEAD.

107. The crosshead is a prolific source of thumping and pounding from various causes, of which the getting loose of the piston rod is one of the most common causes. There are several methods of attaching the piston rod to the crosshead. The rod may pass through the crosshead with a shoulder or taper, or both, on one side of the crosshead and a nut on the other; or the rod may be secured to the crosshead by a cross key, instead of the nut; or the end of the rod may be threaded and screwed into the crosshead, having a check-nut to hold the rod in place. In the first-mentioned case, the nut may work loose, which would cause the crosshead to receive a violent blow, first, by the nut on one side and then by the shoulder or taper on the other at each change of motion of the piston. The remedy is obvious—set up the nut. A similar effect will be produced if the cross key should work loose and back out, the remedy for which is to drive in the key. In the case of the piston rod being screwed into the crosshead and the rod slacking back, the danger is that the piston will strike the rear cylinder head. The check-nut should be closely watched.

108. Another source of pounding at the crosshead is loose wristpin brasses, the remedy for which is to set up on the brasses, but not too tight.

109. In the case of a crosshead working between parallel guides, pounding may be caused by the crosshead being too loose between the guides; in that case the crosshead shoes should be set out.

110. In the case of a slipper crosshead, pounding will result from the wearing down of the shoe, the cure for which is to put a liner between the shoe and the foot of the crosshead or to set it out with whatever means of adjustment are provided.

POUNDING IN AIR PUMP.

111. Pounding in the air pump is generally produced by the slamming of the valves, caused by an undue amount of water in the pump, which will usually relieve itself after a few strokes. The pump piston, however, may be loose on the piston rod or the piston rod may be loose in the crosshead, either of which will cause pounding. A broken valve may also cause pounding in the air pump, all of which must be repaired as soon as detected.

POUNDING IN CIRCULATING PUMP.

112. In a circulating pump of the reciprocating type, pounding may be caused by admitting too little injection water, and the pounding may be stopped by adjusting the injection valve to admit just the right quantity of water. It may so happen, however, that the injection water is very cold, and to admit enough of it to stop the pounding in the circulating pump will make the feedwater too cold. To meet this contingency, should it arise, an air check-valve is often fitted to the circulating pump to admit air into the barrel of the pump as a cushion for the piston; this check-valve may be kept closed, when not needed to admit air, by means of a screw stem above it.

A broken valve, the piston loose on the piston rod, or the piston rod loose in the crosshead will all cause pounding in the circulating pump, the same as in the air pump, and they should all be treated in the same manner as was specified for similar troubles in the air pump.

CONCLUSION.

113. The derangements causing pounding, as well as derangements of machinery in general, produce their own individual sounds, which are easily recognized by the experienced engineer. It is here that the attentive and careful engineer will prove his value, as by taking prompt and judicious action he will prevent a breakdown. He should be able to detect any unusual noise about his engine, though it may be imperceptible to the unpracticed ear. It is almost always the case that any derangement of the parts of an engine will give timely notice by an unusual sound, and if this warning is heeded and promptly acted on by the engineer, a breakdown can generally be prevented. The various sounds produced by an engine while running can be learned only in the engine room by the engineer who is responsible for the proper running of the engine. They cannot be learned in any other way.

114. The engineer, knowing the various causes that produce pounding and thumping in his engine, can prevent them in a great measure by keeping the engine in such good order that they cannot occur.

ENGINE MANAGEMENT

(PART 2.)

HOT BEARINGS.

CAUSE, PREVENTION, AND CURE.

GENERAL INSTRUCTIONS.

1. Introduction.—Hot bearings are the source of much anxiety and annoyance to the engineer, besides interfering very seriously with the proper performance of the engine.

2. Causes.—The primary causes that lead to the heating of bearings may be enumerated as follows:

Newly fitted brasses and journals.
Refitted brasses and journals.
Brasses set up too tightly.
Brasses too loose.
Warped and cracked brasses.
Cut brasses and journals.
Imperfectly fitted brasses.
Brasses pinching the journal at their edges.
Oil feed stopped entirely.
Not enough oil.
Dirty and gritty oils, or oils of bad quality.
Oil squeezed out of the bearings.
Grit from any source in the bearings.

Journals too small, either in diameter or in length.
Overloaded engine.
Engine out of alinement.
External heat.
Brasses fitted too snugly between collars of journal.
Springing of bedplate.
Springing or shifting of pedestal or pillow-block.

3. Best Form of Bearing.—The bearing of an engine in which the shaft journals run should approximate, as nearly as possible, a hole through a solid support. If it were possible, a hole with a bushing of suitable metal in it would form the best possible bearing for a shaft; but since the bearing, however well designed and made, will in course of time wear somewhat, it becomes a necessity that there should be some means of adjusting the brasses, so as to prevent the shaft having a side movement when they are worn.

4. Adjustment of Bearings.—Some engineers consider it an error to make bearings adjustable; they say it gives an opportunity for careless men to do mischief through lack of judgment. It is certainly a fact that one of the principal causes of hot bearings is setting them up too tightly. Some persons, as soon as they hear a pound or noise about an engine immediately conclude that some bearing is slack and tighten it up; this propensity is to be deplored. There are numerous other causes of pounding in engines besides slack bearings, and the engineer should be fully convinced that the pound is caused by slack brasses before setting them up. Bearings on an engine that is in line and in good order, if properly adjusted, will run smoothly and noiselessly for months without having to be touched with hammer or wrench, and it should be the object of an engineer to get his engine into that condition as soon as possible and to keep it so.

5. Watching Bearings.—Bearings, particularly those of large engines, require constant watching. The engineer or oiler should know at all times the condition of every bearing and oil cup; this will require frequent trips around the engine to examine the oil cups to ascertain if they

are feeding and if they contain sufficient oil and to replenish the oil in the cups whenever necessary. While making his rounds, he should feel with the palm of his hand the brasses of those bearings that have shown a tendency to heat and those that are most liable to heat, particularly the crankpins.

TREATMENT OF HOT BEARINGS.

6. Mixtures for Reducing Friction.—Should any of the bearings show an inclination to heat, as indicated by its temperature rising above blood heat or above the temperature of the surrounding atmosphere, the oil feed should be increased; if the oil does not feed freely, run a wire through the oil tubes. If the bearing continues to get hotter, mix some flake graphite (black lead), flour sulphur, or powdered soapstone with the oil and feed the mixture into the bearing through the oil holes, between the brasses, or wherever else it can be forced in. A little aqua ammonia introduced into a hot bearing will sometimes check heating by converting the oil into soap by saponification, soap being an excellent lubricant. Mineral oils will not saponify.

7. Danger of Increasing Heating.—If, after trying the remedies just mentioned, the bearing continues to grow hotter, say to the extent of scorching the hand or burning the oil, it indicates that the brasses have been expanded by the heat and that they are gripping the journal harder and harder the hotter they get; at this stage, if the engine is not stopped or if the heating is not checked, the condition of the bearing will continue to grow worse as long as the engine is running, and may become so bad as to slow down and eventually stop the engine by excessive friction. By this time the brasses and journal are badly cut and in bad condition generally, and the engine must be laid up for repairs.

8. Remedies for Increasing Heating.—The state of affairs mentioned in Art. **7** should not be permitted to be reached. After the simple remedies given in Art. **6** have been tried and failed to produce the desired result, the

engine should be stopped and the cap nuts or key of the hot bearing should be slacked back and the engine allowed to stand until the bearing has cooled off. If necessity requires it, the cooling may be hastened by pouring cold water upon the bearing, though this is objectionable, as it may cause the brasses to warp or crack by unequal contraction. Putting water on a very hot bearing should be resorted to only in an emergency, that is, when an engine *must* be kept running regardless of a spoiled pair of brasses. Water may be used on a moderately hot bearing without doing very much harm. It is quite common in practice, when sprinklers are fitted to an engine, to run a light spray of water on the crankpins when they show a tendency to heat, with very beneficial results.

9. If the engine is not started again until the faulty bearing has become perfectly cool, the cap nuts or key should be set up a little, but not too much, before starting; otherwise, the brasses, having been slacked off, may be too loose, and excessive thumping and pounding will ensue.

10. Dangerous Heating.—Should a bearing become so hot as to scorch the hand or to burn oil before it is discovered or through the necessity of keeping the engine running from some cause, it is imperative that the engine should be stopped, at least long enough to loosen up the brasses, even though it is necessary to start up again immediately, otherwise the brasses will be damaged beyond repair and deep grooves cut into the journals. If the brasses are babbitted, the white metal will melt out of the bearing at this stage. The engine is now disabled, and if there is not a spare set of brasses on hand, it will be inoperative until the old brasses are rebabbitted, if they are worth it, or until a new set is made and fitted. If an attempt is made to rebabbitt a brass while it is in place under the shaft, the chances are that the attempt will result in a failure.

11. Keeping Engine With Hot Bearing Running. If it is absolutely necessary in an emergency to keep the

engine running at all hazards while a bearing is very hot, the engineer must exercise his best judgment as to how he shall proceed. After slacking off the brasses, about the best he can do is deluge the inside of the bearing with a mixture of oil and graphite, sulphur, soapstone, etc., and the outside with cold water from buckets, sprinklers, or hose, taking the chances of ruining the brasses and submitting to cutting the journal. Of course, the engine must be stopped as soon as the emergency has passed and the journal then stripped. It is to be expected that the journal will be found to be deeply grooved and the brasses cut and warped. If the brasses were babbitted, most of the white metal will have disappeared and little else but the framework of the brasses will be left. But if the brasses are made of solid composition or bronze, they can be refitted for at least temporary use or until new ones can be procured.

12. Refitting a Cut Bearing.—The wearing surfaces of the brasses and journal must be smoothed off as well as circumstances will permit; but if the grooves are very deeply cut, it will be useless to attempt to work them out entirely, and if the brasses are very much warped or badly cracked, it will be best to put in the spare ones if any are on hand. If not, the old ones must be refitted and used until a new set can be procured, which should be done as soon as possible. As for the journal, it is permanently damaged; temporary repairs can be made by smoothing down the journal and brasses; but at the first opportunity the journal should be turned in a lathe and the brasses properly refitted or be replaced with new ones.

13. After a bearing has once been heated up sufficiently to cut the brasses and journal or to warp or crack the brasses, it is afterwards constantly in danger of heating up again on the slightest provocation; and the engine is thereby rendered unreliable and uncertain in regard to its steady running. No precaution that can be taken to prevent the heating of bearings is too great to be used for the attainment of this end.

CAUSES OF HOT BEARINGS IN DETAIL.

NEWLY FITTED BRASSES AND JOURNALS.

14. Cause of Friction.—The bearings of new engines are particularly liable to heat, due to the wearing surfaces of the brasses and journal having just been machined. Newly worked metal, when viewed through a powerful microscope, presents the appearance of being a mass of fine needle points projecting outwards. When the newly worked surfaces of two pieces of metal are rubbed together under pressure, the needle points of one piece engage with the needle points of the other piece and excessive friction is produced, the result being that the surfaces in contact are cut into grooves, which still further increases the friction; but if the rubbing process is continued in a moderate manner, so that the surfaces in contact do not cut, the needle points will be bent over gradually, each point forming a small hook. Millions of these little hooks side by side form a shell or a hard surface on the rubbing parts, and the needle points can no longer engage with each other, thereby lessening very greatly the danger of heating by friction and eliminating it entirely when properly lubricated.

15. Wearing Down Bearings.—The conditions mentioned in Art. **14** exist with new brasses and the journal of an engine bearing; therefore, if a new engine or one with new brasses is run moderately, in regard to both speed and load, and with rather loose brasses, until the needle points are bent over, there will be little danger of the bearings heating thereafter from this cause if proper attention is given to their adjustment and lubrication. This is what is familiarly termed **wearing down the bearings.** The impression generally conveyed by this expression is that the metal of the brasses and journal is actually worn away; such is not the case, however, as has been explained. If the journal is true and if the brasses are properly fitted to it, there is no necessity for them to be *worn* down; to bend over the needle points is all that is required.

16. Uneven Bearing of Brasses.—Another source of heating of bearings of new engines is the following: For practical reasons there must be a little play between the brasses and their beds; this permits a slight movement of the brasses when pressure is exerted on them by the shaft; and notwithstanding the fact that they may have been most carefully fitted in the shop, they require a certain amount of running to properly adjust and accommodate themselves to their surroundings. This is especially the case with the bearings of large engines, and the same conditions will obtain every time the brasses are removed. It seems almost impossible in practice to put the brasses of a large bearing back again just where they were before removal; it always requires time for them to settle into their old places; therefore, they should not be disturbed unless there is a positive necessity for doing so. The direct cause of the tendency to heat in this instance is that the brasses do not bear evenly on the journal after the several parts of the bearing are assembled. When a bearing runs well, it is not good practice to disturb it; it is better to leave well enough alone.

REFITTED BRASSES AND JOURNALS.

17. The bearings of an engine that has just been thoroughly overhauled and the journals and brasses of which have been refitted are liable to heat. The wearing surfaces of the bearings having been newly worked or machined, the surface of the metal is in the needle stage, and, also, the brasses have not yet had a chance to adjust themselves to the journal and their beds. The engine, therefore, is in about the same condition as a new engine, so far as the bearings are concerned, and should be treated in the same manner, i. e., it should be run moderately, with loose brasses, until the needle points are bent over and a shell has been formed on the wearing surfaces, and until the brasses have accommodated themselves to their surroundings.

BRASSES SET UP TOO TIGHTLY.

18. When the brasses of an engine bearing are set up too tightly, heating is inevitable, and probably more hot bearings result from this cause than any other, and with less excuse. It is often the case that an attempt is made to stop a thump or a pound in an engine by setting up the brasses when the thump could and should be stopped in some other way.

19. The direct cause of heating of bearings when the brasses are set up too tightly is the abnormal friction that is produced by the brasses binding on the journal. The prevention and cure are obvious. The brasses should not be set up too tightly, and if they are, they should be slacked off as soon as possible. As a matter of fact, hot bearings should never occur from this cause. Only a responsible person should have charge of the bearings and no one else should be permitted to meddle with their adjustment.

BRASSES TOO LOOSE.

20. Bearings may heat on account of the brasses being too loose. The heating is caused by the hammering of the journal against the brasses when the crankpin is passing the dead centers. This derangement is easily remedied, however, by setting up the cap nuts or key. Here the experience and judgment of the engineer is called into play to decide just how much to set up, as it is very easy to overdo the matter and set up too far, with a hot bearing as the result.

21. Most practical engineers have their own particular views regarding the setting up of bearings. One method is to set up the cap nuts or key nearly solid and then slack them back half way; if the brasses are still too loose, they are set up again and slacked back less than before, repeating this operation until the ideal position is reached, that is, when there is neither thumping nor heating. It is important that this desired point be approached very gradually and

carefully, else the chances are that it will be overreached and the operation will have to be repeated all over again.

22. Another method of setting up journal brasses is as follows: Fill up the spaces between the brasses with thin metal liners, say from 18 to 22 Birmingham wire gauge in thickness, and a few paper liners for fine adjustment; put in enough of them to cause the brasses to set rather loosely on the journal when the cap nuts or keys are set up solid. Run the engine for a while in that condition and note the effect; then take out a pair of the liners and set up solid again. Repeat this operation until the brasses have reached the ideal point, when there is neither thumping nor heating, and there let them remain as long as they fill the ideal condition. It may require a week or more, and with a large engine longer, to reach the desired point, but it will be all the better to give the needle points time to be bent over and the brasses time to adjust themselves. If this system of treating bearings is carefully carried out, there will be very little danger of their heating. When the proper point is reached, the engine should run a long time without requiring any further adjustment of the bearings. In removing the liners, great care should be exercised not to disturb the brasses any more than is absolutely necessary. A pair of thin, flat-nosed pliers will be found useful in slipping out the liners. This method is preferable to the first one mentioned, because there is not so much danger of setting the brasses up too far.

WARPED AND CRACKED BRASSES.

23. Warped and cracked brasses will cause heating, because they do not bear evenly on the journal, and hence the friction is not distributed over the entire surface, as it should be. The remedy will depend on the extent of the distortion of the brasses. If the distortion is not too great, the brasses may be refitted to the journal by chipping, filing, and scraping; but if they are twisted so much that they cannot, within reasonable limits, be

refitted, nothing will do but new brasses. Warped and cracked brasses are the result of putting water on them while they are very hot, which should be avoided if possible.

CUT BRASSES AND JOURNALS.

24. Brasses and journals that have been hot enough to be cut and grooved are liable to heat up again any time on account of the undue friction produced by the roughness of the wearing surfaces. As long as the grooves in the journal are parallel and match the grooves in the brasses, the friction is not greatly increased; but if a smooth journal is placed between a set of brasses that are grooved and pressure is applied, the journal crushes the grooves in the brasses and becomes brazed or coated with brass, and then the coefficient of friction becomes very high and heating results.

The way to prevent heating from this cause is to work the grooves out of the journal and brasses by filing and scraping as soon as possible after they occur.

IMPERFECTLY FITTED BRASSES.

25. Faulty workmanship is a common cause of the heating of crankpins, wristpins, and bearings. The brasses in that case do not bear fairly or sit squarely in their beds, and while they appear all right to the eye, they may not be square in the bearing. A crankpin brass must sit squarely on the end of the connecting-rod and the rod itself must be square. If the key, when driven, forces the brasses to one side or the other and twists the strap on the rod, it will draw the brasses slantwise on the pin and make them bear the hardest on one side or the other, thus reducing the area of the wearing surfaces. The same is true of the shaft bearings. If the brasses do not bed fairly on the bottom of the pillow-block casting or do not go down evenly, without springing in any way, they will not run as they should, and heating will result. Chronic heating of bearings is almost always caused by badly fitting brasses. This is a defect that should be looked for and remedied at once, if found to exist.

BRASSES PINCHING THE JOURNAL AT THEIR EDGES.

26. Brasses, when first heated by abnormal friction, tend to expand along the surface in contact with the journal; this would open the brass and make the bore of larger diameter, if it were not prevented by the cooler part near the outside and by the bedplate itself.

If the brass has become hot quickly and excessively, the resistance to expansion produces a permanent set on the layers of metal near the journal, so that on cooling, the brass closes and grips the journal; it will then set up sufficient friction to heat again and expand sufficiently to ease itself from the journal, and so long as that temperature is maintained the journal runs easily in the bearing. This is why some bearings always run a trifle warm and will not work cool. A continuance of heating and cooling will set up a mechanical action at the middle of the brass, which must eventually end in cracking it, just as a piece of sheet metal is broken by continually bending it backwards and forwards about a certain line.

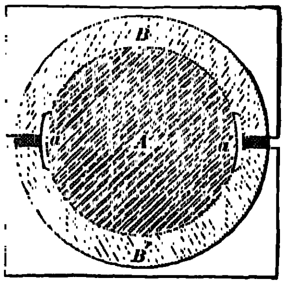


FIG. 1.

27. The cause of heating mentioned in Art. **26** may be prevented by chipping off the brasses at their edges parallel to the journal, as shown at *a* and *a'*, Fig. 1, in which *A* is a sectional view of the journal and *B*, *B'* represent the top and bottom brasses.

OIL FEED STOPPED.

28. It does not take many minutes for a bearing to get very hot if it is deprived of oil. The two principal causes of a bearing becoming dry are an oil cup that has stopped feeding, either by reason of being empty or by being clogged up from dirt in the oil, and oil holes and oil grooves stopped up with accumulated dirt and gum. Both of these conditions are the direct result of negligence, and their existence can always be prevented by the exercise of reasonable care.

NOT ENOUGH OIL.

29. The effects produced upon a bearing by an insufficient oil supply is similar to that of no oil, only in a lesser degree. Of course it will take longer for a bearing to heat with insufficient oil than with none at all, and the engineer has more time in which to discover and remedy the difficulty. As a rule, however, more oil is used on bearings than is actually necessary, and a waste of oil is the result. A drop of oil at the right time and in the right place is just as good as a quart injudiciously applied. A steady feed, a drop at a time, is what a journal requires.

DIRTY AND GRITTY OILS AND OILS OF BAD QUALITY.

30. Oils containing dirt and grit or deficient in lubricating quality are prolific sources of hot bearings; but it is within the province and power of the engineer to guard against such causes. There is a great deal of dirt in lubricating oils of the average quality, as engineers find who strain it; therefore, all oil should be strained through a cloth or filtered, no matter how clear it looks. All oil cups, oil cans, and oil tubes and channels should be thoroughly cleaned out frequently. Oil may be removed from the cups by means of an oil syringe, with which every engine room should be supplied. All oil removed from the cups and cans should be strained or filtered before using. If the above instructions are strictly followed, all danger of bearings heating from the use of dirty and gritty oils will be eliminated.

31. Bearings heating from the use of oils of bad quality are not so easily disposed of, however; there is such a great variety of lubricating oils on the market whose quality cannot be definitively decided upon without an actual trial that it is a difficult matter to avoid getting a bad lot of oil sometimes. About the only safe way to meet this trouble is to pay a fair price to a reputable dealer for oil that is known

to be of good quality, unless the purchaser is an expert in oils. Cheap combination oils, generally speaking, are very deficient in lubricating qualities and hence should be avoided, as also should gummy oils, which choke up the oil channels and glue the brasses and journals together over night.

32. Brasses of very large bearings are often cored out hollow for the circulation of water through them which assists very materially in keeping them cool.

OIL SQUEEZED OUT OF BEARINGS.

33. Bearings carrying very heavy shafts sometimes refuse to take the oil, or if they do it is squeezed out at the ends of the brasses or through the oil holes, when the journal will run dry and heat. The great weight of the shaft causes the journal to hug the bottom brass so closely that the oil cannot penetrate between them, or, if it does, it is immediately rejected. Large journals require oil of a high degree of viscosity, or heavy oil, as it is popularly called. Oil of this character has more difficulty in working its way under a heavy shaft than a thin oil has, but thin oil has not the body necessary to lubricate a large journal.

This difficulty may be met by chipping oil grooves or channels in the brasses. A round-nosed cape chisel, slightly curved, is generally used for this purpose, taking care to smooth off the burrs made by the chisel; a steel scraper or the point of a flat file will do this. The grooves are usually cut into the brass in the form of a **V** if the engine is required to run only in one direction; if it is to run in both directions, the grooves should form an **X**. In the first instance care must be taken that the **V** is forward of the direction of the rotation of the shaft; that is, the grooves should spread out from their junction in the same direction as that in which the journal turns. The oil grooves may be about $\frac{1}{4}$ inch wide and $\frac{1}{8}$ inch deep and semicircular in cross-section.

GRIT IN BEARINGS FROM ANY SOURCE.

34. Grit is an endless and ever-present source of heating of bearings; it is only by persistent effort on the part of the engineer that he can keep his machinery running cool in a dirty atmosphere. Experience is the best instructor in this matter. The causes of this condition are innumerable, therefore, it is only possible to mention a few of them here. The machinery of coal breakers, stone crushers, and kindred industries is especially liable to be affected in this way. Work done on a floor over an engine shakes dirt down upon it at some time or other; all floors over engines should be made absolutely dust-proof by laying paper between the planks to prevent this. A prolific cause of hot bearings from grit, if the engine-room and firerooms communicate, is carelessness in wetting down the ashes and clinkers. If piles of red-hot clinkers and ashes are deluged with buckets of water, which is the common practice, the water is instantly converted into a large volume of steam that rises with a leap, carrying with it large quantities of small particles of ashes and grit that penetrate into every nook and cranny to which it has access, and it will find its way into the bearings sooner or later. Throwing large quantities of water on the hot clinkers and ashes should be stopped; sprinkle them instead and close the fireroom door while the ashes and clinkers are being hauled or wet down or while the fires are being cleaned or hauled.

35. If emery, emery cloth, Bath brick, or other gritty cleaning material is used around a bearing, it is sure to get inside and cause trouble; it is, therefore, better not to use them in too close proximity to a bearing.

36. As a precaution against grit getting into a bearing, all open oil holes should be plugged with wooden plugs or bits of clean cotton waste as soon as possible after the engine is stopped, and should be kept closed until ready to oil the engine again preparatory to starting up. Plaited hemp or cotton gaskets should also be laid over the crevices

between the ends of the brasses and the collars of the journals of every bearing on the engine and kept there while the engine is standing still.

37. Bearings are now in use that, it is claimed by their makers, are dust-proof, but their use does not relieve the engineer from the responsibility of taking every precaution possible to keep grit and dirt out of the bearings of his engine.

JOURNALS THAT ARE TOO SMALL.

38. Journals that have insufficient superficial area of wearing surface will heat. In practice only a certain amount of pressure *per square inch* of area can be sustained by a bearing before the friction reaches the point that will cause heating.

The pressure that a bearing will sustain *per square inch* of area of rubbing surface without heating depends on the materials of which the journal and brasses are composed, the fineness of their finish, the accuracy of their fit the adjustment of the brasses, and the lubricant used.

39. Pressure and friction have a direct relation to each other. Less friction is produced per square inch of surface by a long journal than by a short one of equal diameter with the same total pressure; therefore, a long journal is not nearly so liable to heat as a short one of the same diameter, and a journal of large diameter is not so liable to heat as one of small diameter of equal length. It is the aim of the designer to so proportion the journal that the pressure or friction will not exceed the practical limit that the bearing will sustain. The *total* amount of friction of two bodies in contact depends on the pressure of the one on the other and is nearly independent of the area of the surfaces in contact, hence the necessity of engine journals being large enough to distribute the friction over a sufficient area of surface.

40. There is only one cure for a bearing that heats constantly on account of being too small. This is to make it

larger if circumstances permit it to be done. If this is impossible, the best of lubricant must be used, and if necessary, water must be run constantly on the bearing. It is a good idea to have a set of spare brasses in readiness for an emergency.

OVERLOADED ENGINES.

41. The effect produced by overloading an engine is precisely similar to that of the journals being too small. The pressure on the brasses being increased to a point beyond that for which they were designed, the friction exceeds the practical limit and the bearing heats. The only thing to do to remedy this difficulty is to reduce the load on the engine to within the amount it was intended to stand.

42. In the case of an engine being run at or near its limit of endurance, or if the journals are too small, especially if a large loss should be incurred by the machinery being shut down while new brasses are being made and fitted, it would be a wise and economical precaution to have a complete set of spare brasses, especially if the brasses are babbitted, on hand ready to slip in when the fatal moment arrives, as it surely will.

ENGINE OUT OF LINE.

43. If an engine is not in line, the brasses do not bear fairly upon the journals. This will reduce the area of the wearing surfaces in contact to such an extent that the friction is in excess of the practical limit, which necessarily will cause heating. If the engine is not very greatly out of line, matters may be considerably improved by refitting the brasses by filing and scraping down the parts of the brasses that bear most heavily on the journal. If this does not answer, the heating will continue until the engine is lined up.

44. The crosshead guides of an engine out of line are apt to heat, and they will continue to give trouble until the

defect is remedied. The guides may also heat from other causes; for instance, the gibs may be set up or lined up too much. Of course, if such is the case, they should be slacked off. The danger of guides heating may be very much lessened by chipping zigzag oil grooves in their wearing surfaces and by attaching to the crosshead oil wipers, made of cotton lamp wicking arrauged so as to dip into oil reservoirs at each end of guides if they are horizontal, and at the lower end if they are vertical. These wipers will spread a flim of oil over the guides at every stroke of the crosshead, which will keep them well lubricated.

EXTERNAL HEAT.

45. Bearings may get hot by the application of external heat. This may be the case if the engine is placed too near furnaces or an uncovered boiler, or in an atmosphere heated by uncovered steam pipes or other means. The excessive heat of the atmosphere will then expand the brasses until they nip the journals, which will generate additional heat and cause further expansion of the brasses, and so on until a hot bearing is the result.

46. If the engine is placed close enough to a furnace to cause heating from that source, a tight partition should be put up, if possible; this will also prevent dirt and grit from the fireroom getting into the bearings. If the boilers, steam pipes, and cylinders are unclothed, they should be covered with some good non-conducting material; and possibly a ventilating fan could be rigged up to advantage. Other remedies depend on the conditions and require the judgment of the engineer.

BRASSES TOO LONG.

47. If the brasses are too long and bear against the collars of the journal when cold, they will most surely heat after the engine has been running a while; it is hardly possible to run bearings stone cold, they *will* warm up a little

and the brasses will be expanded thereby, which will cause them to bear still harder against the collars. This, in turn, will induce greater friction and more expansion of the brasses.

48. The evil may be obviated by chipping or filing a little off each end of the brasses until they cease to bear against the collars while running. A little side play is a good thing for another reason, which is that it promotes a better distribution of the oil and prevents the journal and brasses wearing into concentric parallel grooves.

SPRINGING OF BEDPLATE.

49. If the bedplate of an engine is not rigid enough to resist the vibration of the moving parts, or if it is sprung from the uneven setting or the unstability of the foundation, the engine will be thrown out of line either intermittently or permanently, and the bearings will heat from the causes and conditions mentioned in Arts. **43** and **44**; but it will do no good to refit the brasses unless the engine bed is stiffened in some way and leveled up. The form of the bedplate and the surrounding conditions generally must suggest the best way to meet this difficulty.

SPRINGING OR SHIFTING OF PEDESTAL OR PILLOW-BLOCK.

50. The effect of the springing or shifting of the pedestal or pillow-block is similar to the springing of the engine bed; that is, the bearing will be thrown out of line, with the consequent danger of heating. As the pedestal is usually adjustable, it is an easy matter to readjust it, after which the holding-down bolts should be screwed down hard. This is one of the few instances where it is permissible for the engineer to put his strength on the wrench. As a rule, a nut or bolt should be set up just solid; with very rare exceptions, a sledge hammer should never be used in driving a wrench, as 3-inch steel bolts have been broken in this way. It is also very bad practice to drive a nut up with cold

chisel and hammer, unless the nut is in a position that it is impossible to reach it with a wrench.

If a pedestal is not stiff enough to resist the strains upon it and it springs, measures should be taken to stiffen it. The method to be used can only be determined on the spot and calls for the exercise of judgment on the part of the engineer.

LUBRICANTS, LUBRICATION, AND LUBRICATORS.

LUBRICANTS.

INTRODUCTION.

51. Classification.—Lubricants may be divided into three distinct classes, viz., *animal*, *vegetable*, and *mineral*. The name of each designates its origin. There are also lubricants that are composed of a combination of two or more of the above primary classes, which practically form another, or fourth, class.

52. Properties.—The origin of lubricants is not so important to the engineer as their lubricative properties and their power to resist decomposition, vaporization, and combustion by the application of heat; especially is this the case with cylinder oils and valve oils. The value of a lubricant depends on the amount of greasy particles it contains, or its **viscosity**. Other desirable features of a good lubricant are: It should reduce friction to a minimum. It should be free from acids and free alkalies, or, in other words, it should be neutral, and of uniform constituency. It should not become gummy, rancid, or otherwise altered by exposure to the air and it should be odorless. It should stand a low temperature without solidifying or depositing solid matter. It should be entirely free from grit and all

foreign matter. It should be especially adapted to the conditions as to speed and pressure of the rubbing surfaces on which it is to be used; the question of cost is also a consideration. All first-class lubricants possess these properties to a greater or less degree, and each of them is adapted to its own particular class of work. They are also of all degrees of fluidity and solidity—from the thin, light oil used for oiling the indicator down to the thick oils and through the greases to graphite and soapstone for the heaviest journals.

53. Thick or heavy oils are generally considered to rank the highest in viscosity; this is not always the case, however. Some oils of high specific gravity rank lower in viscosity than others of a lower specific gravity, hence the lubricative qualities of an oil cannot always be judged accurately by either its viscosity or specific gravity. Then, again, different manufacturers of lubricants have different standards and names for presumedly the same grade of oil. Furthermore, lubricants that may be very satisfactory for heavy journals might not do at all for light journals, and those that answer well for journals and guides would be very objectionable in cylinders and steam chests—all of which goes to show that different lubricants are required for different purposes.

54. Selection of Lubricants. — Though there are numerous tests for determining the various properties and qualities of lubricants, they, as a rule, involve the use of elaborate chemical apparatus and complicated and delicate machines that are entirely beyond the reach of the average engineer in ordinary engine-room practice. Even the reliability of these elaborate tests is questioned and they are a source of dispute between experts. Under these circumstances it is not an easy task to instruct an inexperienced person how to select a lubricant best suited to his particular needs or to enable him to detect adulterants. Some of the simpler tests will be given further on.

55. In a general way, about the best that an engineer who is not an expert judge of lubricants can do is to procure *from a reputable dealer* several samples of oil or grease that in his judgment are best suited to the machinery he has in charge, taking care to select light-bodied oils for light machinery and to grade his selections down accordingly to suit the size and weight of the journals and the work they have to do. Then he should run the machinery for a stated length of time with each oil, carefully noting the results obtained by each. By the time the engineer has reached the end of his experiments with this assortment of oils, he will have discovered by development and observation which is the best one for his purpose, and it will then only be a matter of common sense to hold on to *that* one until he has good reasons to believe that he can get a better one; then, taking the last one as a standard, he might try another lot of *well-recommended* samples in the same way as before, and so on until he finds the best one for his purpose that the market affords, and at the same time he acquires valuable experience with lubricants. It is important, however, that he should confine his experiments to well-known standard brands of lubricants only, otherwise he will waste much valuable time without gaining a corresponding benefit, and when he finds an oil that, after a fair trial, is satisfactory, he should use it and no other.

56. In selecting the samples for trial, the engineer should examine them very carefully in every possible way and compare one with the other; he should note their color and transparency; rub some of each between the fingers and thumb or on the palm of the hand; note if the sample is smooth and oily and contains no grit; pour a few drops on a sheet of tin or a piece of glass and hold the tin or glass at different angles and note how it flows and if it leaves any residue or gum in its track; examine it with a strong magnifying glass for foreign substances; smell it, and if it is rancid or has a very offensive odor, reject it. If the engineer persists in this practice, it will not take him long to

learn how to distinguish between the different grades and qualities of lubricants, which will enable him to select the one that will best serve his purpose and, at the same time, add very greatly to his general store of engineering knowledge, thereby enhancing the value of his services.

FLUID LUBRICANTS.

57. Animal Oils.—Animal oils are derived from the fats of animals and fish. Those that are used as lubricants are : *Lard oil*, *tallow oil*, *neatsfoot oil*, *horse-fat oil*, *sperm oil*, *whale oil*, *porpoise oil*, *seal oil*, *shark oil.*

Of animal oils, pure lard oil takes the lead as a lubricant for ordinary machinery; but it has the disadvantage of congealing in cold weather; to ward against chilling to a certain extent, the winter strained oil only should be used when it is exposed to a temperature sufficiently low to congeal the ordinary oil.

Tallow oil is very similar to lard oil.

Horse fat is sometimes used in place of tallow, but its odor is offensive.

Clarified neatsfoot oil is an excellent lubricant for light machinery.

Of the fish oils, sperm is the best lubricant, but its scarcity and high price precludes its general use. It is used, principally in a refined state, for oiling indicators and other delicate mechanisms. Whale and porpoise oils are sometimes used in place of sperm oil, but they are inferior to it. Seal and shark's liver oils are used as adulterants. Menhaden fish oil should not be used as a lubricant, as it quickly turns rancid and gums.

58. Vegetable Oils.—Vegetable oils are derived from the fruits, seeds, and nuts of trees and plants. They compare very favorably with animal oils as lubricants, and several of them are excellent for that purpose. The leading vegetable oils that are used for lubricating purposes are: *Olive oil*, *rape-seed oil*, *colza oil*, *cottonseed oil*, *castor oil*, *palm oil.*

The olive oil is probably the leading vegetable oil used for lubricating machinery, but all the others in the above list are fairly good for that purpose. Castor oil and cottonseed oils are more liable to gum than pure olive oil. Linseed oil, either raw or boiled, should not be used as a lubricant; it dries quickly and is very gummy. Cocoanut oil (palm oil) soon becomes rancid and in that condition it is not a good lubricant.

59. Mineral Oils.—Mineral lubricating oils are distilled from bituminous shale and from the residuum of crude petroleum after the volatile oils and illuminating oils have been distilled off at various temperatures up to 572° F. The products of the petroleum still, when heated to temperatures above 572° F., are the lubricating oils. These oils are graded according to their specific gravities and are named as follows:

PROPERTIES OF MINERAL OILS.

No.	Name.	Specific Gravity.	Flashing Point.	Burning Point.
1	Solar oil...........	.860 to .880	370° F.	435° F.
2	Mixed oil..........	.880 to .890		
3	Spindle oil, No. 1 ..	.895 to .900		
4	Spindle oil, No. 2 ..	.900 to .906	394° F.	468° F.
5	Machine oil, No. 1..	.906 to .910	426° F.	487° F.
6	Machine oil, No. 2..	.910 to .915	441° F. and up.	525° F. and up.
7	Cylinder oil, pale...	.915 to .920		
8	Cylinder oil, dark..	.920 to .950		
9	Vulcan oils.........	.910 to .960		

60. Besides the oils given in the table, there are many other mineral lubricating oils on sale under different names, each manufacturer naming his own product to suit himself, but the above list will serve to show the method of grading mineral machine oils in regard to their specific gravities and their flashing and burning points.

All the mineral oils given in the table, if pure, are excellent lubricants, each one being adapted to its specific purpose for light, medium, and heavy machinery and cylinders.

The color of a mineral lubricating oil is not always an indication of its purity or value. A dark-colored oil may be purer than a light-colored one; therefore, in selecting a mineral oil, too much stress should not be laid upon its color.

61. Compounded Oils.—There is a great variety of compounded oils manufactured for all sorts of purposes and at all prices. They are, generally speaking, simply made to sell without regard to merit or value as lubricants. Herein lies the danger of being defrauded in purchasing cheap oils. They are, as a rule, compounded of thin, light oils, which lack the viscosity, or body, for lubrication, and a variety of substances to produce an artificial body that adds nothing to their lubricative properties. Most, if not all, of the adulterants used for this purpose are of a gummy nature and enemies to good lubrication. If mineral oils are used as the bases of these compounded oils, they are liable to have a low flashing point, which renders them totally unfit for use in cylinders. In fact, these oils had better be entirely ignored by the engineer; but as they are made and doctored to imitate the pure standard oils, they are well calculated to deceive the unwary, as it is not an easy matter to detect the difference between them by mere inspection.

62. A trial on the engine is the best method to test the merit of a lubricant, though some simple tests, as described under the heading "Tests of Lubricants," may be made with beneficial results.

63. Economy.—Lubricants, like everything else that is exposed for barter or sale, are worth just about what is paid for them. A good article must always fetch its price and a poor article is sold cheaply.

There is no economy in buying cheap lubricants; they cost less per gallon, but it takes more gallons to do the

required work. Now that excellent oil filters are to be had, enabling the drip oil to be filtered and used over again, there is no necessity for using cheap oils.

64. Greases.—Greases are divided into three classes, viz., *compounded*, "*set*" or *axle*, *boiled* or "*cup.*"

65. Compounded greases are made by mixing cheap oils with fats, paraffin, and the various waxes. They soon become rancid, in which state they are unfit for lubrication, being instead friction producers. It is hardly necessary to say that the engineer should avoid these greases, even though they are cheap.

66. Set, or axle, greases are mixtures of low-grade oils and fats converted into grease by the application of lime. They are cheap greases, used principally for lubricating axles of vehicles and the like, and are familiarly known as **cart grease.** These greases are unfit to use in the bearings of engines.

67. Boiled, or cup, greases are those that are well adapted for engine lubrication. They are produced chemically and are not simply mechanical mixtures as are the others. They are perfectly neutral and will remain so indefinitely. They are made by saponifying fats and fatty oils with lime and dissolving the soap in mineral oil.

68. Soaps made by the use of soda or potash are soluble in water, while soaps made by the use of lime are insoluble in water.

There is a series of greases in this class that are made by saponifying the fats and fatty oils by means of caustic soda; the soaps thus made are soluble in water. These greases are good lubricants if properly made, but they are apt to contain either an excess of alkali or an excess of acid; in either case they are liable to be injurious to the bearings. Free acids or alkalies may be detected by the litmus-paper test.

69. Cup and engine greases include: Nos. 1 to 4 cup greases, Nos. 1 to 3 Albany greases, sponge greases, crank-pin greases, gear greases, lubricating packing, plumbago and graphite greases.

SOLID LUBRICANTS.

70. The solid lubricants are: *graphite*, *soapstone*, *sulphur*, *mica*, *metaline*. These may properly be classed under the general head of mineral lubricants.

71. **Graphite,** called also **plumbago** and **black lead,** is used for lubrication either in the form of a powder, flaked, compressed into bushings, or by being mixed with wood fiber and solidified in molds by pressure; this latter is called **fiber graphite.** After being removed from the molds, the forms are thoroughly dried and then saturated with a drying oil, after which they are exposed to a current of hot dry air to oxidize the oil and to harden the mass. When hard they may be worked the same as metal. Fiber graphite is claimed to be self-lubricating.

72. The powdered and flake graphite are used to mix with greases for heavy journals and also to mix with the ordinary engine oils to cool a hot bearing. When graphite is used as a lubricant, the journal becomes covered with a thin coating of graphite, which reduces friction to a minimum.

73. **Soapstone, sulphur,** and **mica,** in the form of powder, are sometimes mixed with oils and greases to improve their lubricating qualities for heavy and hot journals. Sheets of mica pressed together and held firmly in a casing have been used instead of brasses with fair success.

74. **Metaline** consists of small cylinders of graphite fitted into holes drilled in the surface of the bearing; it is said to require no other lubrication.

LUBRICATION.

75. The object of lubricating the bearings of an engine is to reduce the friction of those parts that rub against one another to a minimum and to prevent the rubbing surfaces becoming hot, which, if the rubbing is continued without lubrication, will ultimately cause seizing, thereby permanently damaging the bearings and rendering the engine inoperative. The lubricant attains its object by interposing itself in the form of a thin film between the rubbing surfaces, either by gravity or pressure, and thus prevents the rubbing surfaces coming into direct contact with one another.

76. Animal and vegetables oils have been used as lubricants for many years, but since the introduction of multiple-expansion engines and high steam pressures, mineral oils have come into very general use, especially for lubricating pistons and slide valves, for the reason that mineral lubricating oils are not carbonized by high-pressure steam as readily as are animal or vegetable oils. Moreover, animal and vegetable oils (called **fatty** oils to distinguish them from mineral, or **hydrocarbon,** oils) are decomposed by the great heat of high-pressure steam and form stearic, palmetic, and oleic acids. These acids when hot readily attack iron, steel, copper, and its alloys; therefore, cylinders, pistons, etc. are eaten away when fatty oils are used for lubricating them.

77. The acids formed by the decomposition of fatty oils are particularly destructive to steam boilers when the exhaust steam is condensed and used as feedwater, as is the case with condensing engines having surface condensers. On the other hand, mineral oils are not affected by alkalies, therefore the old method of saponifying the grease in boilers and surface condensers by boiling them out with soda or potash is ineffectual when mineral oils are used in the cylinders; in that case, if the boiler tubes or condenser tubes become coated with grease, it must be removed by hand. It is far better, however, to keep the grease out of the condenser and boilers entirely by placing an efficient grease extractor in the

exhaust pipe between the low-pressure cylinder and the condenser.

78. High-grade cylinder oils only should be used for lubricating pistons and slide valves, and the flashing point should not be lower than 400° F. The higher the temperature of a hot bearing, the less is the lubricating power of the oil or grease used; consequently, a lubricant that may be thoroughly efficient at ordinary temperatures may be ineffectual in reducing the friction of a bearing that has suddenly become heated; hence the practice of mixing graphite, flour sulphur, etc. with the oil to increase its body and lubricative properties.

TESTS OF LUBRICANTS.

APPARATUS.

79. In giving the following simple tests of lubricants, there is no pretence that they are absolutely accurate in the sense of determining the commercial values of lubricating oils and greases, but they will serve the purposes of the engineer very well and assist him greatly in the selection of his lubricants; besides, they have the merit of being within the reach of every engineer in his ordinary engine-room practice.

80. The pieces of apparatus required to make these tests are few and inexpensive. They consist of an ordinary tin or iron pan 8 or 10 inches in diameter and 3 or 4 inches deep; a metal cup about the size and shape of an ordinary tumbler; a high-grade thermometer that will measure at least 500° F.; a couple of quarts of clean white sand; a half-dozen clear, white glass ½-pint bottles; a large sheet of tin or plate of glass, preferably the latter; a sheet each of red and blue litmus paper; a common thermometer; a quart of gasoline; a few pounds of ice; a pound or two of rock salt and about the same quantity of sal soda (washing soda); a small iron boiler or saucepan; a small quantity of caustic soda or concentrated lye; a pane of glass painted black on one side with

a mixture of shellac varnish and lampblack; and a small tin funnel.

81. Test for Acids and Alkalies.—Dissolve a small quantity, say a teaspoonful, of the oil or grease to be tested in five or six times its bulk of boiling water, in which steep a piece of red litmus paper; if the litmus paper remains red after having been soaked in the mixture for a considerable length of time, the oil or grease is *acid.* If the color of the paper turns to dark blue quickly, the oil is *alkali.* If it changes color very gradually to a light blue, the oil is *neutral.* As a check on the above test, try the mixture with a piece of blue litmus paper in the same way. If the color of the paper does not change, but remains dark blue, the oil is alkali. If the paper turns red quickly, the mixture is acid; but if the paper changes very gradually to a pale red, the solution is neutral.

82. Test for Viscosity.—Pour a few drops of each sample of oil upon the large sheet of tin or glass while the sheet is perfectly level, then raise one end of the sheet gently about 1 inch and support it in that position; watch the race of the drops of oil down the inclined plane. The oil that reaches the bottom of the plane *last* ranks highest in viscosity. Of course, this is only a comparative test, but it will enable the operator to select the oil best adapted to his purpose from a number of samples. After making a selection, it would be well to try the precipitation test given in Art. **87** on it for artificial viscosity.

83. Greases cannot be tested for viscosity in the way described in Art. **82**; about the only convenient method for the engineer to do this is by rubbing some of the grease between his fingers and thumb or in the palm of the hand, noting the result. After some practice he will be able to judge approximately the viscosity of the sample.

84. Flashing and Burning Tests.—Pour some of the oil that is to be tested into the metal cup until it is nearly full; place the cup in the pan and surround the cup with

sand until the pan is filled with it; place the pan and contents on a hot stove, over a gas jet, or in any other convenient place for heating it; immerse the bulb end of the high-grade thermometer in the oil in the cup and watch the rise in temperature; when it reaches 300° pass a lighted match slowly across the top of the cup; repeat this every two or three degrees rise in temperature until the vapor arising from the oil ignites with a flash, then note the temperature as indicated by the thermometer; it is the **flashing point.** Continue the test until the oil ignites and burns on the surface. When that occurs the reading of the thermometer gives the **burning point.**

85. The Cold Test.—Partly fill the metal cup with a sample of oil; place the cup in the pan; fill the pan around the cup with cracked ice mixed with rock salt and sal soda; cover the apparatus over with a piece of bagging or blanket and keep it covered until the oil in the cup is congealed; then remove the freezing mixture from the pan and fill the pan with hot water; when the oil in the cup commences to melt, immerse the bulb of a thermometer into it and note the temperature; it is the **congealing point.**

86. Saponification Test.—If it is desired to ascertain if animal or vegetable oils are mixed with oil that is represented to be pure mineral oil, it may be determined as follows: Place about a pint of the oil into the small iron boiler or saucepan and add 1 or 2 ounces of caustic soda or concentrated lye; boil the mixture for ½ hour and then set it aside to cool. A tablespoonful of chloride of sodium (common salt) thrown into the mixture while cooling will hasten the process. When thoroughly cool, examine the mixture; if the surface is covered with soap, the oil contains animal or vegetable fats; otherwise it is pure mineral oil.

87. Precipitation Test.—The precipitation test is for the purpose of ascertaining if the oil contains paraffin, waxes, gums, etc. Place an ounce of each of the oils in a separate ½-pint bottle, pour 2 ounces of gasoline into each bottle on top of the oil, and shake the bottles until the oil is dissolved

by the gasoline; then allow the mixtures to settle. If there is any considerable amount of precipitation or sediment in any of the bottles, it indicates that the oil in them has been treated to produce artificial viscosity and should be rejected.

88. Test for Mineral Oil Mixed With Fatty Oils.—The presence of mineral oil when mixed with animal or vegetable oils may be detected by pouring a drop of the suspected oil upon the sheet of blackened glass and holding the glass at various angles to the light; if it shows rainbow colors, it contains mineral oil.

89. Test to Detect Sulphur in Mineral Oils.—Heat a small portion of the oil to 300° F. in the metal cup and pan of sand and maintain that temperature for about 15 minutes; after cooling, if the sample is considerably darker in color than the original oil, it is unfit to use in cylinders or on hot bearings.

AUTOMATIC LUBRICATORS.

CLASSIFICATION.

90. The devices used for the automatic lubrication of steam engines and similar machinery may, in accordance with their purpose, be divided into two general classes, *bearing lubricators* and *steam lubricators*.

91. A **bearing lubricator** may be defined as one intended for, and only applicable to, the lubrication of bearings. This class is divided into three subclasses, *plain* and *sight-feed* lubricators, and *grease cups*. **Plain** and **sight-feed** bearing lubricators are intended and can only be used for oil; **grease cups,** as implied by the name, are built to use grease.

92. Steam lubricators are intended for the lubrication of the moving parts in contact with the steam; they may be

divided into *mechanical*, *water-displacement*, and *hydrostatic* lubricators. A **mechanical steam lubricator** generally has the form of a force pump; it may be operated by hand, in which case its action is intermittent. A hand-operated mechanical steam lubricator is generally fitted only as an emergency device, to be used when the automatic lubricator is out of order. When a mechanical lubricator is operated continuously by some moving part of the engine, its action is automatic. **Water-displacement lubricators** depend for their action on condensation of steam in the reservoir containing the oil; the latter being lighter than water floats on top and overflows into a suitable passage as the water in the bottom of the reservoir increases. **Hydrostatic lubricators** depend for their operation on the pressure generated by a head of water furnished by condensation of steam.

BEARING LUBRICATORS.

93. A **plain lubricator** is the simplest form of a device for automatic lubrication; it generally takes the form shown in Fig. 2. It consists of a body *a* fitted with a central tube *b* and a removable cover *c*. The oil contained in the body *a* is led into the central tube by capillary attraction, a few strands of lamp wick, as *d*, carrying the oil over. The advantage of this oiling device is its simplicity; the disadvantages are its unreliability and its lack of adjustment of the oil feed. The latter can be adjusted to some degree by changing the number of strands of lamp wick; as the flow of oil is not in plain sight, however, there is always some doubt about the action of the lubricator.

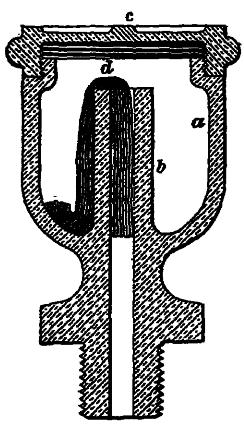


FIG. 2.

94. A **sight-feed bearing lubricator,** as implied by the name, has the oil feed in plain sight. The oil generally is fed by gravity, flowing through an annular opening in the base of the lubricator. The general appearance of this device is shown in Fig. 3. It consists of a glass oil reservoir *a* having a central tube *b* with a valve seat inside of it and at its lower end. A valve *c*, which can be locked in any position by the locknut *d*, serves to regulate the flow of the oil. The oil enters through the hole shown in the lower end of the tube *b*. The drops of oil issuing from the tube *b* show plainly in the sight-feed glass *e*. The upper cover has a hole in it through which the reservoir is filled; a movable cover *f* serves to keep out the dust.

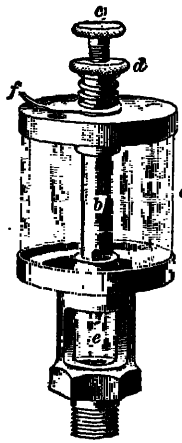


FIG. 3.

95. Various attachments are used for conveying the oil from a stationary sight-feed lubricator to the moving parts, as the crankpin, eccentric, and wristpin. Fig. 4 shows how the oil may be carried to the crankpin by a so-called centrifugal oiling device. The oil from the lubricator *a* flows through the pipe *b* into the ring *c*, which connects to a hole drilled in the center of the crankpin through the fixture *d* that is fastened to the crankpin. The oil entering at *c* passes to the crankpin by the centrifugal force generated by the revolution of the crank and through radial holes out of the crankpin between the surface of the crankpin and the brasses. The main bearing simply carries the stationary lubricator *e*, which discharges directly into the bearing. A separate lubricator *f* may be fitted for the eccentric, discharging into a long trough or funnel *g* fastened to the eccentric strap.

96. Fig. 5 will serve as a suggestion of how automatic lubrication of the wristpin and guides may be obtained. To lubricate the upper guide, the stationary lubricator *a* is used;

a lubricator *b* is placed at a sufficient distance above the level of the lower guide to cause the oil to flow through the channels shown to the guide. To lubricate the wristpin from a stationary cup *c*, a wiping device *d* is attached to the wristpin. This carries the wiper *e*, which is adjusted so as

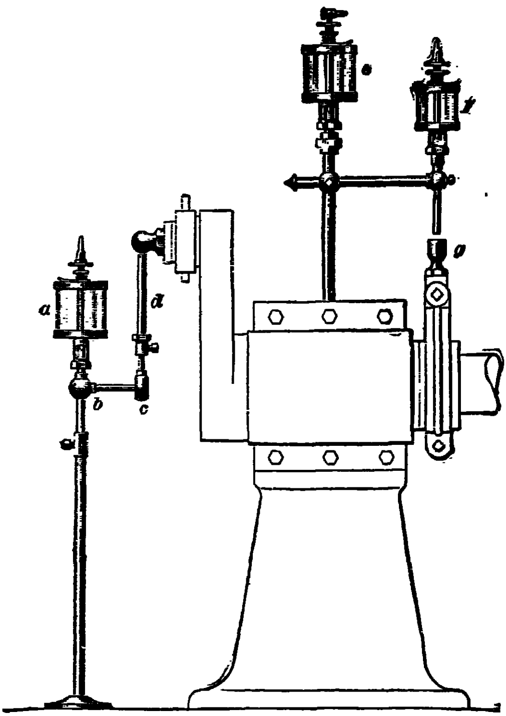


FIG. 4.

to wipe off the drops of oil hanging at the bottom of the nozzle *f* as the crosshead passes back and forth. The oil thus collected flows by gravity through a hole in the center of the wristpin and is delivered through one or more radial holes to the outside of the pin.

A precisely similar wiping fixture may be and often is used for crankpins and eccentrics, using stationary lubricators placed on top of the main bearing.

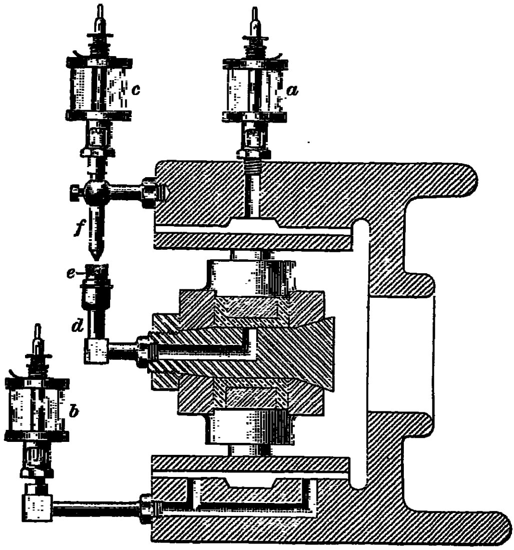


FIG. 5.

97. **Grease cups** are made in various ways and either as **plain** or **compression** cups. In a plain cup the grease only flows down by gravity as the heat of the bearing melts it; to assist the grease, it is a good practice to put a piece of small copper wire in the hole through which the grease leaves the cup. A compression grease cup may be hand-operated or spring-operated; Fig. 6 shows one of the type first named. By screwing the cap down by hand over the base, the grease is forced out.

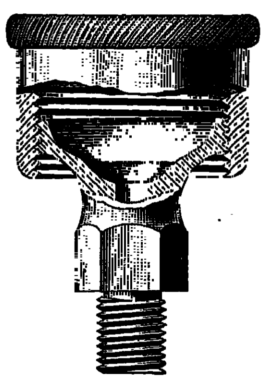


FIG. 6.

98. Spring-operated compression grease cups have a piston, on top of which is placed a spring that continually forces out the grease. In most of them the rate of flow can be regulated by a suitable valve.

STEAM LUBRICATORS.

99. Mechanical Lubricators.—Hand-operated mechanical steam lubricators are generally small force pumps connected to a suitable oil reservoir and having the discharge pipe connected to the main steam pipe close to the throttle. Their construction and operation is so simple as to require no description.

100. Automatic mechanical lubricators are operated from some moving part of the engine, as some convenient part of the valve gear. Fig. 7 shows one form of such a device, known as the "Rochester automatic lubricator." It consists of a cylindrical oil reservoir *a* containing a piston that is screwed down on the oil by gearing connected to the ratchet wheel *b*. The ratchet wheel is operated by a pawl on one end of the ratchet lever *c*, which is vibrated back and forth by the rocker *d*. This rocker is rocked back and forth by some convenient reciprocating part of the engine. The connection between *c* and *d* is made in such a manner that the arc through which *c* vibrates can be changed so that the pawl will move the ratchet wheel any desired number of teeth within the range of the device. The oil is ejected from the reservoir by the piston and passes through *e* to the engine.

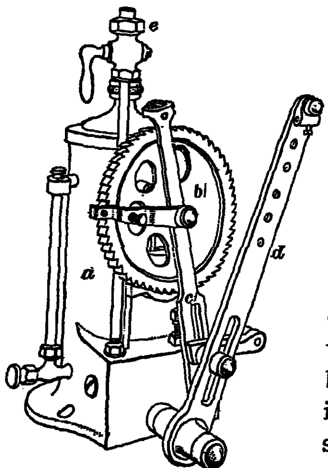


FIG. 7.

101. Water-Displacement Lubricators. — The simplest form of a water-displacement lubricator is shown in Fig. 8. It consists of a cylindrical shell *A* provided with a central tube *a*; a cap *C*, through which the lubricator is filled; and a shank *b* for attaching it in a vertical position

to the steam chest or steam pipe. A valve *B* controls the communication between the lubricator and the engine.

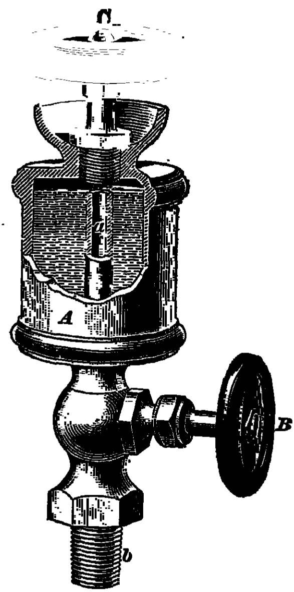


FIG. 8.

102. The operation of the lubricator is as follows: The receptacle is filled with oil and closed. The valve *B* is then opened, thus allowing the steam to pass through the central tube in to the top of the lubricator. The steam, coming in contact with the cold surfaces of the oil and receptacle, condenses. Since water is heavier than oil, bulk for bulk, the drops of condensed steam sink to the bottom of the receptacle. As two bodies cannot occupy the same space at the same time, the drops of water displace a quantity of oil equal in volume to their own; the oil, which has no other means of egress, flows over the edges of the central tube and runs by gravity into the steam pipe.

The objectionable features of this lubricator are that the flow of oil is not readily controlled and that there is no indication of when the lubricator stops working, either for want of oil or otherwise.

103. To overcome the objections mentioned in Art. **102**, **sight-feed** water-displacement lubricators have been designed, one of which is shown in Fig. 9. Its principle of action is the same as that of the lubricator shown in Fig. 8; i. e., it depends on the condensation of the steam and the subsequent displacement of the oil. Its construction is as follows: A cylindrical receptacle *d* is provided with a central tube *a* communicating with the threaded shank *e* and the sight-feed glass *A*. To fill the receptacle,

the cap *E* is provided. The upper end of the lubricator communicates with the sight-feed glass by the passage *b*. In operation the steam is admitted to the lubricator by means of the valve *B*, the opening of which admits it to the inside of the lubricator as well as to the sight-feed glass *A*. The steam, coming in contact with the oil and the top of the lubricator, condenses and displaces the oil, which then flows through the passage *b* into a conical nozzle, as shown, and issues from the latter either drop by drop or in a thin stream, depending on the position of the regulating valve *C*. It is apparent that by screwing down the latter, the annular opening between the valve and nozzle is reduced, and hence the flow of oil is checked. Conversely, by screwing up the valve, the rate of flow is increased. The drops of oil issuing from the nozzle flow by gravity through the passage *c* and thus to their destination. Since the glass tube is transparent, the oil dropping from the nozzle is in plain sight of the attendant. By means of a drain cock, not shown in the figure, the lubricator may be emptied when required. This lubricator uses a **down feed**, which means that the oil is discharged downwards in respect to the feed nozzle. These lubricators are not very reliable in their action, since the oil is not forced through the feed nozzle, but only flows through it by gravity.

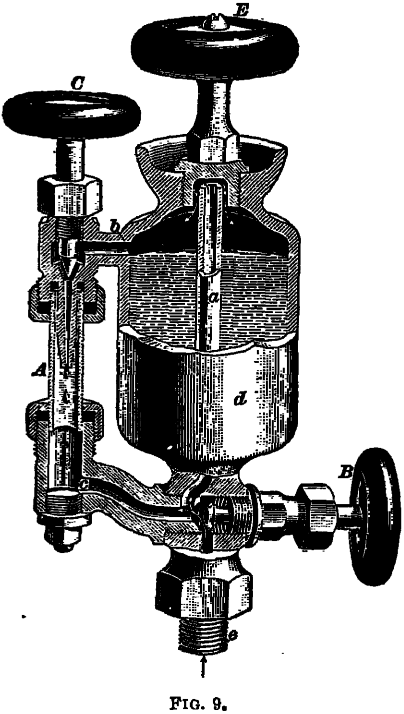


FIG. 9.

104. Hydrostatic Lubricators.—All water-displacement lubricators belong to the **single-connection** type, this meaning that there is only one connection to the steam pipe and, consequently, that the oil must pass through the same passage through which the steam is admitted. Hydrostatic lubricators are made in two styles, *single-connection* and *double-connection*. In a **double-connection** lubricator there are two connecting pipes to the steam pipe, the steam being admitted through one pipe and the oil leaving the lubricator through the other.

105. A typical single-connection hydrostatic lubricator is shown in Fig. 10, (*a*) being a part section and (*b*) a side view. The lubricator is connected to the steam pipe through the nipple *M*. The steam flows through *M* and the pipe *B* into the **condenser** *F*; it also flows through the connection *b* and a passage cored out in *C* to the sight-feed glass *H*. The steam is condensed, both in the condenser and in the sight-feed glass, by radiation. The water in the condenser flows through the pipe *1* into the bottom of the oil reservoir and forces the oil to the top, exerting a hydrostatic pressure on the bottom of the oil, which is transmitted through the oil. The latter flows through the pipe *2* into a nozzle located in the bottom of the sight-feed glass and out of the nozzle into the glass. The drops of oil ascend, by reason of oil being lighter than water, to the top of the sight-feed glass, which, it will be remembered, is filled with water. The oil then flows into the passage within *C* and passes through *b* into the nipple *M* and into the steam pipe.

106. There is an equal steam pressure on top of the water in the condenser and in the sight-feed glass, so that the pressure impelling the oil out of the lubricator is only that due to the hydrostatic head. The rate of flow of the oil through the nozzle in the bottom of the sight-feed glass can be regulated by means of the needle valve *E*; the water can be shut off from the oil reservoir *A* by closing the valve *D*; a drain cock *G* is used for draining the reservoir. A gauge glass *g* shows the amount of oil in the lubricator.

The reservoir can be filled when the filling plug *O* is unscrewed. A small valve *S* is closed in order to shut off the steam from the sight-feed glass in case the latter is broken or in need of cleaning. With the valves *D* and *S* shut, the gland in which the valve *E* works is unscrewed;

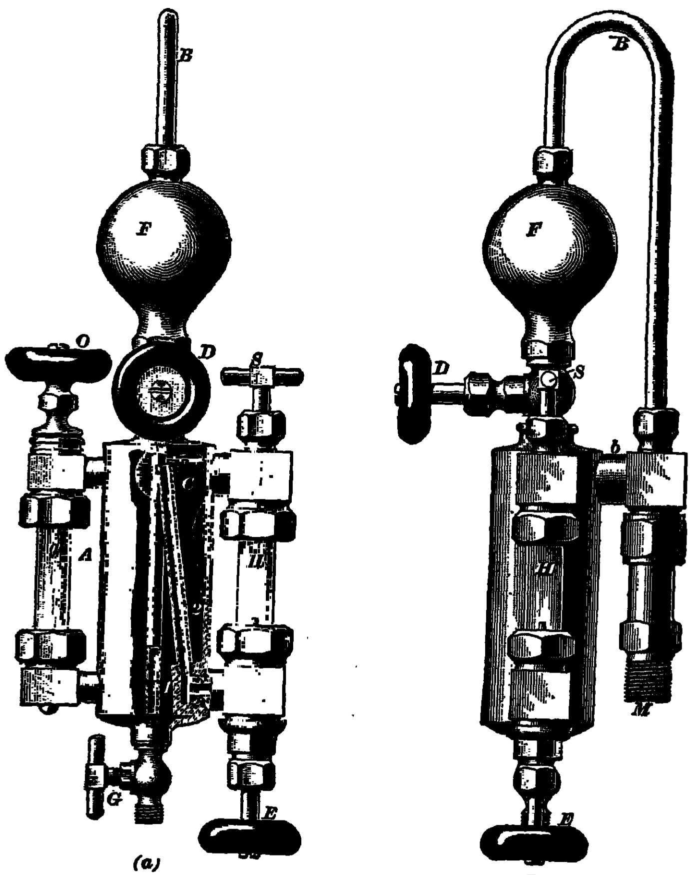


FIG. 10.

the broken or dirty glass tube can be removed and a new one or the cleaned old tube inserted. It will be noticed that this lubricator has an up feed; that is, the drops of oil coming from the nozzle flow upwards in the sight-feed glass.

107. To start the lubricator, open the valves *D* and *S*; to stop it, close the valves *D* and *S*. The regulating valve *E* when once adjusted need rarely be disturbed. To drain the lubricator while steam is on the pipe into which it delivers, close the valve *D*, and *E* and *S* being open, open *G*. When not under steam, to drain remove the filler plug *O* and open *G*.

108. A **double-connection lubricator** is shown in Fig. 11, which incidentally shows the mode of attachment to a vertical pipe. The condenser *a* has its independent steam connection; the angle valve *b* admits the steam to the condenser. The water in the condenser passes to the bottom of the reservoir *c* through the pipe *d* and forces the oil upwards into the pipe *e* leading to the bottom of the sight-feed glass *f*. It then flows through an annular opening regulated by the valve *g* up the sight-feed glass and through the pipe *h* and valve *i* into the steam pipe. The pressure impelling the oil forwards is simply the hydrostatic pressure due to the water in the condenser.

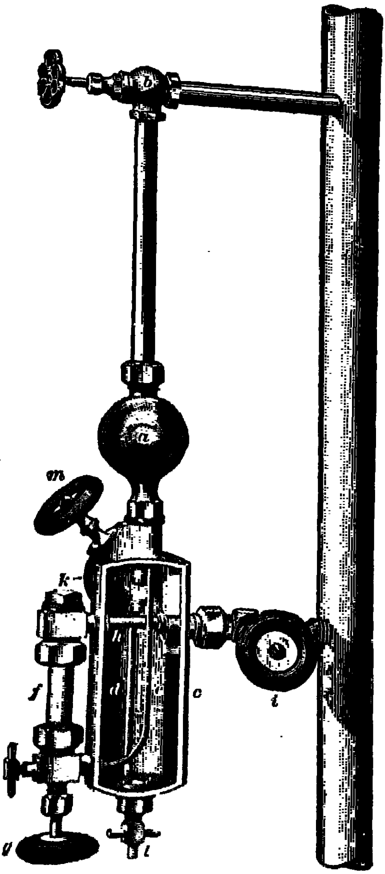


FIG. 11.

109. To fill the lubricator, close the valve *k*, which shuts the condenser off from the reservoir *c*, and also close the valve *g*. Open drain valve *l* and

remove filler plug *m*. When water has drained off, close valve *l*, fill with oil, and replace the filler plug. Open valve *k* again and regulate the flow with the valve *g*. To shut off the lubricator temporarily, close the valve *k*; to shut it off permanently, close valves *b* and *i*.

110. Double-connection lubricators are made in which the condenser is an independent vessel; such a one is shown connected up to a horizontal overhead steam pipe in Fig. 12. The receptacle *o*, which may be filled by unscrewing the cap *m*, communicates with the sight-feed glass *c*. The regulating valve *i* controls the flow of the oil. At *j* a valve used for draining the receptacle is shown; the drain pipe may be attached by the union *k*. The valve *l* may be used for closing the passage leading from the bottom of the lubricator to the pipe *a*. The lubricator is connected to the steam pipe *n* by the pipe *a*, which connects *o* to the condenser *p*, which is, in turn, connected to *n* by the pipe *f*. The oil from the lubricator passes to the steam pipe through the pipe *b*. By means of the valves *g* and *h*, the lubricator may be shut off when desired. Its operation is as follows: When starting the cup for the first time, the pipes *a* and *b* and the sight-feed glass *c* are filled with water, the pipe *a* being filled nearly up to *d*. Since the water in the pipe *a* can flow into the bottom of the lubricator, it follows that the oil will

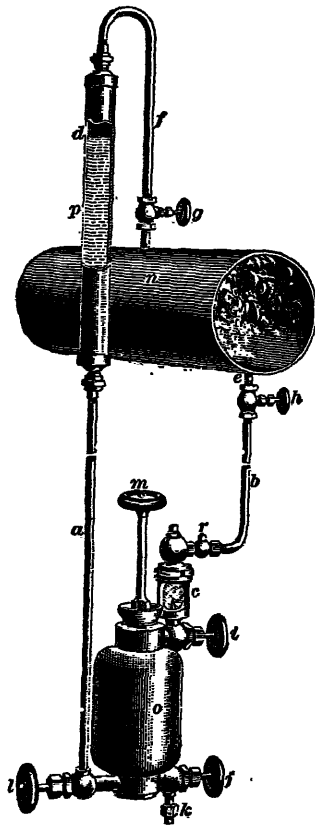


FIG. 12.

be forced through the feed nozzle with a pressure depending on the hydrostatic head *de*.

After passing through the feed nozzle, the drops of oil ascend through the sight-feed glass and up the pipe *b*, the pressure causing the upward flow being due to the difference in specific gravities of the water and oil. To prevent the emptying of the pipe *b* when draining the lubricator preparatory to replenishing the oil supply, a small check-valve *r* is provided. In order to replenish the water that passes from the pipe *a* into the lubricator, the condenser *p* is used. This may be a vessel of any desired shape; it is usually a piece of 1½-inch brass tubing, as shown in the figure. The steam entering from the steam pipe is condensed by coming into contact with the relatively cool surfaces of the condenser; the latter is made large in order to increase the radiating surface. In this style of lubricator the hydrostatic pressure operating the device may be made as great as circumstances will permit by simply extending the loop of the pipe *f* higher up. If this is done, the condenser must also be raised in order to derive the most benefit from the change.

111. Double-connection lubricators should never have one connection attached to the steam pipe between the throttle and boiler and the other between the throttle and engine. If the lubricator is connected in this manner, upon closing the throttle there will be full steam pressure on the condenser and none on the sight-feed glass. In consequence, the lubricator will very rapidly be emptied, the steam pressure forcing all the oil out into the engine. If circumstances require the connection to be made in this manner, a special locomotive double-connection sight-feed lubricator should be selected. Such a lubricator is especially made in such a manner that the oil cannot leave the reservoir when the throttle is closed.

ENGINE INSTALLATION

COMPARISON OF TYPES OF RECIPROCATING ENGINES.

VERTICAL VERSUS HORIZONTAL ENGINES.

1. Uses of Vertical Engines.—The inverted vertical engine—that is, the engine with the crank-shaft resting in a bedplate placed on the foundation and suitable and appropriate housings containing the guides, the cylinder resting upon and secured to the housings or engine frame—is now the prevailing type for large power-station purposes and many other applications of steam engines. That type of vertical engine in which the cylinder joins the bedplate and has the shaft or beam on top of the engine framing is only used for special and peculiar applications, such as slow-speed pumping and blowing engines, but never for quick-running engines.

2. Controlling Features.—The vertical engine has its distinct application, its advantages, and its disadvantages. The two controlling features that dictate the use of the vertical engine are (1) available floor space for the engine and (2) size of engine. The first reason is self-evident. As to the second reason, in very large horizontal engines, and particularly with the low-pressure cylinder of compound engines, the problem of supporting the weight and preventing the cutting of massive low-pressure pistons running at

the speed now common becomes one of great magnitude; in fact, the success is always problematical, even with the most carefully planned constructions. This bad feature of the horizontal engine is entirely overcome by making the engine vertical; the weight of the piston is then borne by the shaft bearings.

3. Supporting Pistons of Horizontal Engines.—Many devices have been tried to support the weight of large pistons, such as tailrods having crossheads running on external guides, but the distance between the points of support or crossheads is usually long, and the allowable deflection can rarely exceed $\frac{1}{32}$ inch, so that this expedient, to be of any service whatever, requires very large rods. Take the case of a 72-inch cylinder having a 72-inch stroke. With a carefully designed cast-iron piston, it would require a piston rod at least 14 inches in diameter having a 5-inch hole through it to support the piston successfully. Pistons having a steam pocket underneath them, into which steam is admitted through a small hollow tailrod, have been used by one very large builder. Forged steel-plate pistons having broad composition shoes riveted to the lower circumference, the shoes projecting into recesses formed in the heads, have been used by an English builder; while very broad pistons in which the weight of the piston does not exceed 3 pounds per square inch of projected area are often resorted to. Many of the devices for supporting pistons have merit, but many engineers believe the vertical engine to be the best solution of the problem.

4. Inaccessibility of Vertical Engines.—The vertical engine is much more inaccessible than the horizontal machine for oiling, inspecting, and repairing; indeed, in some of the very latest American high-grade engine designs, it would be necessary to dismantle the whole machine to remove the crank-shaft, although the specifications usually demand that it be possible to remove the crank-shaft bearings when the shaft is raised $\frac{1}{4}$ inch. The vertical engine costs on an average about 12 per cent. more than the

horizontal engine. Generally, the vertical engine will not receive the same degree of care and attention that the horizontal machine will, owing to its inaccessibility and the labor and exertion required to reach its various parts. This should not be the case, but it is so, nevertheless.

5. Comparison of Headroom.—The vertical engine requires quite a high building, not only on account of the design, but also because extra room is needed to draw out the piston and piston rod. If the engine is large, a substantial crane or other means of handling the various parts is necessary. The horizontal engine, where space is available and other conditions do not preclude its adoption, has many practical and commercial advantages over the vertical engine. It is the cheaper engine, is much more accessible for repairs, oiling, and inspection, and can be cared for by men physically incapacitated to handle a vertical engine.

6. Comparison of Floor Space.—The horizontal engine from the nature of its design requires considerable floor space, and in localities where property is valuable, as is often the case along city water fronts or at the center of a large electrical distributing system, and where it is desirable to concentrate as much motive power in as small a space as possible, the horizontal engine must give way to the more expensive and less accessible vertical engine. In very large power plants it is quite customary either to connect the engine galleries, making them continuous throughout the whole plant, or to construct mezzanine galleries around the house on the same level and connecting with all the galleries. If this is done, the various units can be visited for inspection and oiling without descending to the floor each time, thus making easier the labor of attending to this class of engine.

7. Influence of Drainage.—Unless especial care is exercised in the design of the vertical engine to free of water all parts coming in contact with the live steam and to prevent water pockets, its economy will fall below that of the horizontal engine, especially if the latter is of the four-valve

type, which type when embodied in the horizontal machine is almost self-draining. The vertical design does not so readily lend itself to drainage and hence requires especial care in this respect.

8. Influence of Balancing.—The mechanical efficiency of the vertical engine is from 2 to 3 per cent. higher than that of the horizontal engine. With an equal measure of care as regards balancing, the vertical engine will operate more smoothly than the horizontal machine; this is due to the fact that the unbalanced vertical force acts vertically through the machine and foundation, while the unbalanced horizontal force is close to the foundation and is counteracted by two heavy masses—the foundation below and the engine above.

9. Combined Vertical and Horizontal Engines.—A type of engine occasionally used is a combination of the horizontal and vertical machine. This engine is usually made a compound, in which the low-pressure cylinder is made vertical, for reasons that have been previously given, while the high-pressure cylinder is placed horizontal. Both engines act on one crankpin, thus making a compact machine having all the advantages of two cranks at right angles. This type of machine has been made in very large units and used for direct-connected electric service and reversing rolling-mill service.

AUTOMATIC CUT-OFF VERSUS THROTTLING ENGINES.

10. Types of Automatic Cut-Off Engines.—There are two distinct types of automatic cut-off engines, the *positive automatic cut-off*, in which the main or cut-off valve closes the port by positive motion derived from some part of the engine, usually an eccentric on the main shaft, and the *releasing-gear cut-off*, in which the main valve or cut-off valve is made to close the admission port by means of vacuum pots, weights, springs, steam pressure, or other sufficient means.

11. Limiting Speeds of Releasing-Gear Engines.—The positive automatic cut-off engine is always used for quick-running engines in preference to the releasing-gear engine. The practicable maximum speed of the latter type may be set at 100 revolutions per minute for engines up to 500 horsepower, 85 revolutions per minute up to 2,000 horsepower, and 75 revolutions per minute up to 5,000 horsepower, although small releasing-gear engines have been run at 150 revolutions per minute. The best builders do not advise speeds higher than those given above. There are a very few builders that run their small and medium-size engines about 20 per cent. faster than the speeds given above, but this does not mean that the same power is obtained at 20 per cent. less investment, for the reason that to run satisfactorily at these high speeds the machines must be especially constructed and heavily built; furthermore, the lack of insurance against shut down due to breakage or heating and the larger quantity of oil required more than offset the item of first cost. For all conditions requiring high speeds and great economy, the positive automatic cut-off engine is usually chosen. These machines can always be run at speeds not determined or limited by the construction or operation of the valve gear.

12. Economy of Automatic Cut-Off Engines.—The economy of the positive automatic cut-off engine with one valve is only about 75 per cent. of that of a releasing-gear automatic cut-off engine of equal grade. There are, however, some positive automatic cut-off engines of the four-valve type in which the cut-off valve is mounted on the main valve and is positively driven by a shaft governor; in such an engine the economy of steam is fully equal to that of the releasing-gear engine. They are capable of much higher speeds than the releasing-gear engine, but owing to some complication of the valve gear, they are not usually run at as high speeds as the one-valve positive automatic cut-off engine. The four-valve positive automatic cut-off engine is somewhat more expensive than the

releasing-gear engine, which, in turn, is considerably more expensive than the one-valve automatic engine.

13. Accessibility of Releasing-Gear Engines.—The releasing-gear engine is invariably more complicated than the positive gear and requires closer adjustment, but on the other hand, it is much more accessible for adjustment, even while in motion, than the positive gear, which is unapproachable while the engine is running. The problem of oiling the positive-gear engine is one that cannot be solved too carefully, as the success of the gear largely depends upon the perfection of the oiling devices. The oiling of the releasing-gear engine is an easy problem in comparison. These statements apply not only to the valve gear, but with equal force to the reciprocating and rotating parts of both classes of engines.

14. Comparison of Throttling and Automatic Cut-Off Engines.—The simple throttling engine is the oldest type of engine and is probably the least used at the present time. Its strongest claim to existence is simplicity, and for many purposes and locations the claim is strong. It is much less economical than the automatic cut-off engine, is usually built for slower speeds, and generally there is not much attention paid to the features conducive to economy. One of the defects of the throttling engine is that, for the purposes of regulation, this machine must have from 5 to 20 pounds more steam pressure on the inlet side of the governor than on the outlet side; consequently, the boiler must generate steam from 5 to 20 pounds higher pressure than is actually used in the engine; hence, some waste of heat takes place before the steam arrives in the working cylinder.

15. The automatic cut-off engine adjusts its energy to the resistance by measuring out a supply of steam always at or near the boiler pressure and sufficient to overcome the resistance; the throttling engine always supplies the same volume, but varies the pressure to suit the resistance to be overcome. The automatic cut-off engine is capable of

high ratios of expansion; the throttling engine as usually built is not, $\frac{3}{4}$ cut-off being the prevailing point, giving only $1\frac{1}{3}$ expansions.

16. There is another class of engine, known as the *Meyer valve engine*, belonging to the throttling-engine family, in which a separate expansion valve on the back of the main valve and worked by a separate eccentric is used to effect a cut-off from 0 to $\frac{3}{4}$ stroke. This class of engines is capable of high ratios of expansion and hence is quite economical; they are usually well made and provided with a throttling governor. The point of cut-off is adjustable by hand and is set very close to the actual demands, allowing the governor very little range of pressure to adjust the speed of the engine.

SIMPLE ENGINES VERSUS COMPOUND.

17. Influence of Power Required.—Like most engineering problems, the problems relating to the use of compound engines resolve themselves chiefly into problems of finance. The cost of fuel and amount of power required are leading factors in determining the use of compound engines, generally speaking. When the power required is less than 200 horsepower, it will hardly pay to put in a compound condensing engine where the steam pressure is limited to 100 pounds gauge, unless the cost of fuel is very high, say $4, or more, per ton. If 125 pounds of steam can be carried by the boiler, it will be a paying investment. Similar limits apply to the case of a compound non-condensing engine, except that the steam pressures should be changed to 125 pounds gauge pressure and 150 pounds, respectively.

18. As the size of the engine increases, it becomes more important to compound, for the reason that a 1,000-horsepower engine does not cost five times as much as a 200-horsepower engine of similar design and construction. When the price of fuel is low, compounding becomes of less

importance, and compound non-condensing engines with a variable load when working with steam pressure not more than 150 pounds are rarely paying investments.

19. Triple- and Quadruple-Expansion Engines.—Triple-expansion condensing engines have shown a real economical advantage over compound engines of 20 per cent. Such engines should not be used with less than 160 pounds steam pressure. Triple-expansion non-condensing engines seldom prove a good investment under ordinary conditions, and the same may be said of quadruple-expansion condensing engines. This statement refers to land engines, but not to marine engines, where quadruple engines are sometimes used not only to secure extreme economy in the use of steam, but also to reduce the vibrations of the ship to a minimum.

20. Steam Consumption.—A good four-valve automatic cut-off engine will consume 24 pounds of dry steam at 100 pounds pressure per horsepower per hour, while a compound condensing engine of similar design, but having a reheating receiver supplied with 50 square feet of tube reheating surface for each cubic foot of steam delivered from the high-pressure cylinder, will consume but 14 pounds of dry steam of 135 pounds pressure per horsepower per hour. A good triple-expansion condensing engine, if supplied with steam at 160 pounds pressure, could accomplish the same work with 11 pounds of dry steam per hour. The above figures as to steam consumption hold only for medium and large engines, say from 500 horsepower up; small engines are not so economical as large engines, which is probably due to the greater ratio of cylinder and port surface to the volume swept through by the piston in small engines.

21. Factors to be Considered.—While the problem of simple versus compound engines is always one of finance, economy of fuel and first cost are not always the determining elements. The compound engine is always more complicated and hence more liable to a breakdown, and if

isolated, requires the carrying of more spare parts. It requires a higher degree of skill to maintain it in economical condition and requires better and more expensive boilers, but does not require as many boilers or as large a boiler plant. The question of whether insurance risks may be greater and the facilities for repairs should also be considered in determining the type of engine. The real test in any case is the final influence of the machinery used on the profits of the business.

TANDEM COMPOUND VERSUS CROSS-COMPOUND ENGINES.

22. Cylinder Arrangements of Tandem Compound Engines.—A tandem compound engine is one in which the cylinders are arranged one behind the other, both pistons being on the same piston rod and acting on one crankpin. The cylinders are arranged sometimes with the high-pressure cylinder behind and sometimes with the low-pressure cylinder behind. Both arrangements have their advantages. When the low-pressure cylinder is placed behind and the front low-pressure cylinder head is made on an internal flange, the cylinders and pistons are quite accessible. When the high-pressure cylinder is placed behind the low-pressure, the piston rod must be removed through the front low-pressure stuffingbox and, consequently, can have no projecting collars forged on it to take the thrust of the low-pressure piston. The rod is sometimes fitted with loose steel collars that take the thrust of the low-pressure piston; sometimes the portion of the rod that enters the low-pressure cylinder is made rather large in order to form a sufficient shoulder for the low-pressure piston to bear against. When it becomes necessary to forge a collar on the rod to secure a sufficient bearing shoulder for the low-pressure piston, the stuffingbox throat must be bushed; the collar will then pass through the throat when the bushing is removed. Both pistons should have tapered seats on the rod.

23. Disadvantages of Tandem Compound Engines. The principal objection to the tandem engine is the inaccessibility of the cylinders and pistons for inspection or repair. The cylinders are also liable to get out of alinement if not properly designed and constructed, which occurrence, in turn, reduces the mechanical efficiency of the machine. The loss of alinement is obviated to a considerable extent by making a heavy cast-iron sole plate extend under both cylinders. The front cylinder should be securely bolted to this sole plate, while the rear cylinder should be arranged to slide in suitable ways, which constrain it laterally, but allow it to move longitudinally when it expands and contracts. This feature in large engines is important.

24. Comparison of Spare Parts Required.—The economical performance of the two types of machines are the same. The cross-compound engine has considerably more parts than the tandem, but many of them are exact duplicates, so that in isolated districts the cross-compound engine would probably not require a larger item of spare parts than the tandem. On account of their smaller size, for equal engine power, the first cost of spare parts for a cross-compound engine would be less than for the tandem engine.

25. Comparison of Mechanical Efficiency.—For equal engine power, the mechanical efficiency of the two types of machines should be in favor of the cross-compound engine. This at first thought seems erroneous, but a little consideration will make this fact clear. The frictional resistances of pistons and rods should be the same with both, but in the tandem compound they are much more liable to increase in time, due to its greater liability to get out of alinement. The valve-gear resistances should be practically the same in both types, but usually are slightly in favor of the tandem compound. Considering the resistances at the crossheads and crankpins, while there are twice as many parts in contact in the cross-compound engine, the total force and resultant pressures and the direction and duration of the pressure are the same in both types.

26. The greatest divergency in the frictional resistances of the two types of engine is at the shaft. For equal degrees of unsteadiness of rotation, both engines working at the same economical ratio of expansion and speed, the tandem engine requires a wheel about $1\frac{6}{10}$ times heavier than a cross-compound engine. This, in turn, requires a larger and heavier shaft and bearings and means an increased velocity of the bearing surfaces, and hence more wear and oil; in this respect the cross-compound engine has a decided advantage over the tandem.

27. Comparison of Cost.—The tandem engine has its strongest claim in the matter of first cost; if this, however, is carefully investigated, it will be found that for similar service, economy, speed, pressure, and type, the first cost of the tandem engine will average only about 9 per cent. lower than the first cost of the cross-compound engine. The cost of foundation for a tandem engine will be about 20 per cent less than that for the cross-compound.

28. Formerly, it was the practice to make the passage of steam from the low-pressure cylinder to the high-pressure cylinder of tandem engines as short and direct as possible, but the prevailing practice at present for equal duty is to give the tandem engine a reheating receiver of a volume equal to that of the receiver of the cross-compound engine, which is usually equal to the volume of the low-pressure cylinder; and it is customary to provide for both types of engines about 50 square feet of tube reheating surface for each cubic foot of steam exhausted by the high-pressure cylinder. Formerly, there was considerable difference of cost between receivers and piping for tandem and cross-compound engines, but as at present constructed, there is no appreciable difference.

29. Comparison of Smoothness of Running.—With equal elaboration to secure smoothness of running, and comparing condensing engines, the tandem engine will generally excel. The reason for this is seen when it is considered that compression is the main factor tending to secure smoothness

in turning the dead centers. If the vacuum in the low-pressure cylinder be good, the remaining gas is so attenuated that ordinary means will not secure sufficient compression to absorb the inertia of the reciprocating parts at the end of the stroke, the result being a severe pounding at all journals. To prevent this, extraordinary and expensive means must be used, such as providing separate valve gear to drive the exhaust valves independent of the steam valves. In the tandem engine, both pistons being on the same rod, sufficient compression can easily be obtained behind the high-pressure and low-pressure pistons to fully absorb the inertia of the reciprocating parts. This applies particularly to releasing valve-gear engines.

SINGLE VERSUS DUPLEX ENGINES.

30. Purpose of Duplex Reversible Engines.—A duplex engine consists of two simple engines, usually exact duplicates in all respects, operating on one crank-shaft; hence, it is similar in arrangement to the cross-compound engine. Reversible engines are almost invariably duplex to facilitate starting the engines at any possible position at which the cranks may happen to be. Familiar examples of this type of engine are the locomotive, hoisting engines, blooming engines, and barring engines.

31. Purpose of Duplex Non-Reversible Engines.—The duplex non-reversible engine is frequently met with in industrial works, and their existence is usually due to an extension of the industry, where a little forethought has served to save the additional cost of an entirely new engine. In planning and developing an industry, it is reasonably expected to grow and expand; often the exact expansion cannot be predicted with certainty. While a certain amount of surplus power can be provided for in installing the original engine, it is a well-established fact in steam engineering that an underloaded engine is an extremely wasteful and poor paying investment; this fact creates the field for the

duplex non-reversible engine. The wheel for the original single engine is made sufficiently large to transmit double the original power, if belt or rope transmission is used; this, however, does not mean that it shall be double the weight or cost, but only 1.4 times the weight for a single engine and about 1.3 times the cost of a wheel for a single engine. Frequently a section of the bedplate containing the shaft bearing is purchased with the original machine, and when the demand for another engine is made, it can be readily attached to the original machine without a shut-down or delay of the works.

32. Methods of Providing for Increased Power.— Other methods are sometimes practiced to accomplish the ends secured by the duplex engine, such as purchasing a larger engine than is required, inserting a thick bushing in the cylinder, and providing a smaller piston, all being so arranged as to be removed when the demand for increased power is made. While this accomplishes the same result, it is done at a sacrifice of economy and ties up a considerable amount of extra capital, due to the first cost of the larger engine, which might otherwise be turned to good investment.

If conditions so change between the time of installing the original engine and such time as increased power is required, the machine intended to be duplex can quite as readily be made into a cross-compound as a duplex by adding a low-pressure cylinder and receiver, and if water be available, a condenser. It might be well to add here that if it is required to obtain the same amount of power with a higher degree of economy, the steam pressure and speed remaining the same, it cannot be obtained by the addition of a high-pressure cylinder, as is very often erroneously assumed. The saving in fuel by compounding and the addition of a condenser should be between 15 and 20 per cent.

33. Comparison of Mechanical Efficiencies. — The mechanical efficiency of the simple engine of equal power compared with the duplex should be a little higher. There should be no appreciable difference in the economical

performance of the two types of engines unless the sizes are such as to render the duplex very small engines, in which event the economical efficiency of the duplex engine will suffer a loss. The duplex engine should operate more smoothly than the simple because of the more even turning moment at the crank-shaft. The simple engine of equal power, and if run at the same speed as the duplex engine, to secure the same coefficient of unsteadiness of rotation, will require a wheel 1.6 times heavier than the duplex, and necessarily a heavier shaft and larger bearings; this operates to reduce the mechanical efficiency of the simple engine.

34. First Cost.—For engines of equal power, under the same steam pressure and piston speed, the duplex engine will cost about 1.4 times as much as the simple engine, while the foundations will cost about 1.6 times as much as the foundations for a simple engine. It is to be noted, however, in selecting engines with reference to cost per horsepower that the price will be found to vary on either side of a minimum horsepower, which for ordinary engines will be about 500 horsepower. Owing to this fact, it sometimes happens that a very large duplex engine may be found to cost less than a single engine of equal power.

HIGH-SPEED VERSUS SLOW-SPEED ENGINES.

35. Classification.—The line of demarcation between high-speed and slow-speed engines is not clearly defined when referred to the number of revolutions made per minute or per second, for there is another characteristic that, when considered in connection with the number of revolutions per minute, assigns them to the class in which they belong. This characteristic is the manner in which regulation is secured. Engines with a releasing-valve gear, usually of the four-valve type, and regulated by a pendulum governor are classed as slow-speed engines, barring a few exceptions, while positive automatic cut-off engines,

regulated by means of a shaft governor, are invariably classed as high-speed engines. Engines regulated by means of a throttling governor are seldom classed as either high-speed or slow-speed engines; in fact, they usually run at speeds midway between those of the two first-mentioned classes of machine. In considering the relative merits of high-speed and slow-speed engines, the speed refers to the revolutions per minute, and *not to the piston speed*, for, as a matter of fact, the piston speed of modern slow-speed engines often exceeds that of the high-speed engine.

36. Purpose and Advantages of High-Speed Engines.—It may safely be said that the advent of electricity as a medium for the transmission of energy caused the development of the quick-running engine, and it is in the electrical field that it still finds its largest market. It is not confined to the electrical industry, however, and has been applied to nearly every service, either direct-connected or by belting. Its principal merits are comparative low first cost and small space required; its principal objections are wastefulness of fuel and need of constant attention. These objections are not universal, however, as there are some high-speed engines on the market that are quite equal in economy and in all other respects to any of the slow-speed engines. There are also some high-speed engines of the enclosed crank-chamber type that, to a great extent, are self-lubricating and demand very little attention; they usually take the vertical form. The majority of high-speed engines are not very economical machines and must be carefully watched.

37. Comparison of Valves for High-Speed Engines. High-speed engines are almost invariably fitted with a balanced valve, which is frequently a piston valve. It is claimed that this type of valve, if used on a horizontal high-speed engine, will begin to leak about the time the engine is paid for and will not improve with age, notwithstanding the many devices used to adjust the fit of these valves in their liners or casings.

The piston valve applied to the vertical engine has given better results as regards less leakage and resultant economy, but even here it has not been altogether satisfactory, principally because the system of regulation imposes varying travel to the valve and unequal wear on the internal surface of the casing or liner. Other systems of balancing are by means of pressure or cover-plates; these require very careful design and workmanship, but if properly designed and fitted, they are much superior to the piston valve. The clearance with the latter form of valve is usually much less than with the piston valve, but the clearance is generally large in all of them, and it is due to this fact that the periods of compression can be lengthened and the engine be made to operate very smoothly.

The Corliss valve has been used to some extent on high-speed engines, but the result has not been altogether encouraging, and in several instances they have been absolute failures, which was probably due to the fact that if pressure is allowed to remain on Corliss valves sufficiently long to force out the film of oil that is between the valve and the seat, it takes considerable force to move them.

38. Valve Motions.—The valve motion is invariably derived from an eccentric of variable throw and angular advance or from an equivalent crank, so hung as to give a nearly constant lead. The peculiar valve gears of the high-grade slow-speed engine, by virtue of which the valves move but little and very slowly after they have closed the ports, are seldom or never attempted in the high-speed engine.

39. Effects of Large Clearance Volume.—The necessary simplicity and desirable low first cost of high-speed engines has resulted, on account of the types of valves, in large and comparatively long steam ports, which increase the clearance volume and clearance surface. As was previously pointed out, high-speed engines have a high speed as regards the number of revolutions per unit of time; consequently,

the stroke must be shortened, which results in a high percentage of clearance volume; the frequency with which this clearance volume or part of it is filled with fresh steam affects the economy of this class of engine to some extent.

40. Effect of High Speed on Regulation.—Aside from the question of economy, one of the leading characteristics of high-speed engines is the regulation. On account of the high rotative speed, the regulator or governor has a much greater opportunity to effect changes in the speed; that is, if the high-speed engine is running 300 revolutions per minute while the slow-speed engine is making 100 revolutions per minute, the high-speed engine may be said to have 600 opportunities to adjust the steam supply while the slow-speed machine has only 200; or, in a unit of time, which may be taken as 1 revolution of the slow-speed engine, the high-speed engine has had 3 opportunities to adjust its speed. Consequently, the regulation of the high-speed engine is much superior to that of slow-running engines, even though they be fitted with governors equally sensitive. As a matter of fact, the better types of high-speed engines, as at present constructed, leave nothing to be desired in the matter of regulation for any possible commercial service.

41. Prevention of Accidents.—The increased risk of wear and the liability to accident due to their rapid motion, and especially when accidents do occur, the seriousness of their nature must be considered in connection with the high-speed engine. The prevailing tendency among builders of this class of engine is to reduce the possibility of accident by selecting higher grades of material, providing liberal wearing surfaces, which are case-hardened or oil-tempered, and using safe and thoroughly tested constructions embraced by massive and well-distributed framings.

42. Lubrication.—High-speed engines require copious lubrication, and unless careful provision be made to collect

the excess, great wastes may result in this direction. This is provided against to a great extent by providing splashers, oil guards, drip pans, and in some designs completely enclosing the running parts in oil casings; in some systems provisions are made for draining and collecting all oil in a separate chamber, where it is carefully strained or filtered and automatically returned to the bearings. In this so-called **return system,** a liberal stream or several streams of oil are kept running upon the bearing surfaces.

43. Accessibility of High-Speed Engines.—From the compact, rigid nature of the design of high-speed engines, they are not as accessible as the slow-running machine, but it cannot be argued that they are particularly difficult of access.

44. Influence of High Speed on Weight of Flywheel. Owing to the velocity of the high-speed engine and to the fact that the energy of a flywheel increases as the square of the velocity of the center of inertia, the wheels for high-speed engines can be made very much lighter and still obtain the same degree of unsteadiness as in the slow-running engine. This relieves the bearings of much dead weight and allows the shaft to be made smaller and makes the velocity of its rubbing surfaces much less.

45. Comparison of Economic Performances.—One of the elements in high-speed engines that, no doubt, contributes much to the economy of the machine is the little time allowed for initial condensation of each charge of steam and for the changes in temperature preceding each charge; some of the single-acting very quick-running engines have met with not a little success, their designers attributing it to the fact above mentioned. It must be borne in mind that the very highest duties and efficiencies have been obtained from the slowest running engines, as pumping engines, and many engineers contend that speed is not of vital importance in securing high economy. There is,

however, little basis for comparison between the two engines, for slow-speed engines can also be denominated as high-grade engines, while high-speed engines may be classed as low-grade engines; and while there may be no appreciable difference in the economical performance of high-grade engines when run at varying speeds, the economy of a high-speed engine would fall away materially if run at a slow speed.

46. Savings Due to High Speed.—The high-speed engine has it strongest claim over the slow-speed engine in its adaptability to direct-connected work, whether the connection be to electric generators, the shaft of a mill, or any industrial work. There is at once a direct saving not only in the first cost of the engine, but in saving due to the omission of transmission machinery, as jack-shafts, belts, or gearing, bearings and their foundations, and the continuing expense resulting from their attendance, lubrication, and repair. High-speed engines, owing to their greater steam consumption, demand a 20-per cent. larger boiler plant, which is an item of first cost to be considered. While the circulars of high-speed engine builders announce their capacity and willingness to build this type of machine for large powers, they are seldom met with in actual practice; the common range of power is from 60 to 200 horsepower, but they are occasionally built in units as large as 800 to 1,000 horsepower.

47. High-Speed Compound Engines.—High-speed engines are as commonly built compound and triple-expansion as are slow-speed engines, and they are more frequently built compound non-condensing than are slow-speed engines. The compounds are arranged both cross and tandem. With high-speed engines it is quite important to have spare parts on hand.

48. Economy of Slow-Speed Engines.—The slow-speed engine, which at the present time is almost invariably

of the four-valve drop cut-off or releasing-valve gear type, is most commonly chosen for all large units where continuous operation is required. In localities where fuel is expensive, even though the steam plant be used as a relay in case of failing water-power, the slow-running economical engine will be found. This condition is generally one requiring very careful study to obtain maximum commercial efficiency. The slow-speed engine admits of many, though practical, complications to the end of securing extreme economy of steam and high mechanical efficiency. The valves are usually so placed as to reduce the clearance volume and clearance surface to a minimum; great care is exercised to free the cylinders of water; steam jacketing of heads and cylinders is common. The polishing of the internal faces of heads and pistons in order to reduce the activity of the metal in receiving and imparting heat to the working steam, and thus reducing initial condensation, is sometimes resorted to; elaborate valve gears to give theoretical steam distribution are possible with the slow-speed engine.

49. First Cost of Slow-Speed Engines.—The slow-speed engine is much the larger, heavier, and more expensive machine, and usually costs $1\frac{3}{10}$ times as much as the high-speed engine of equal power. The foundations are also more expensive, but the boiler capacity need not be so large. The relative complete cost of high-speed and slow-speed power plants is not far from $50 per horsepower for high-speed and $70 per horsepower for slow-speed plants, the engines being simple non-condensing. The economical performance, assuming the engines to be in fairly good condition, should be about 30 pounds of water per horsepower per hour for the high-speed and 24 pounds of water per horsepower per hour for the slow-speed, bearing in mind, however, that the slow-speed engine will for a long time maintain its economical performance, while the high-speed engine will generally lose in efficiency.

50. Direct-Connected Engines for Dynamos.—The prevailing practice in the generation of electricity by steam power is towards direct-connected units, and the cost of the electrical generator is an item constantly urging higher rotative speeds; for this reason the slow-speed engine, a few years ago, was by several builders forced past its practical limits of speed. The impracticability of the move, as made apparent by constant breakage and short life, was soon recognized, and moderate speeds of about 90 to 100 or 110 revolutions per minute were returned to. These speeds are now seldom exceeded, and somewhat slower speeds are advised when long life and immunity from accidents are desired. Reducing the rotative speed of direct-connected engines operates to increase the diameter of the armature or revolving field of the electrical generator, and this, in turn, increases the first cost of the unit; it is here that the high-speed engine obtains its grip in the solution of the problem; but after passing a very moderate power, say 200 to 300 horsepower, the item of economy becomes of such magnitude that the more economical and more expensive slow-running unit establishes its ultimate value to the engineer or purchaser.

51. Attention and Workmanship.—Where the high-speed engine has its disadvantages, corresponding advantages may be found in the slow-running engine, and vice versa; slow-speed engines besides being much more economical do not require the close attention demanded of high-speed engines. With reasonable attention the slow-speed engine will give fair warning in many instances of approaching danger that will be developed too quickly in the high-speed engine to control. The slow-speed engine is more accessible and more readily adjusted; the workmanship need not be so exacting for equal results, the rate of mechanical depreciation is less, and the life and service of the machine is greater. The regulation of the slow-speed engine is not equal to that of the high-speed engine, and they require very heavy wheels for the same degree of unsteadiness.

SELECTION OF ENGINES.

INTRODUCTION.

52. The problem of selecting an engine for a specified service is one demanding all the skill, experience, and forethought of the constructing engineer, for on it rests one of the vital and constant items of expenditure of the industry of which it forms so important a part. The elements determining the selection of an engine may be briefly mentioned here as the influence of the kind of service, the location, first cost, cost of fuel delivered at the boilers, steam pressure available, the duration of service, the facilities for repairs, the kind of labor available, the existing conditions, if additions or renewals, whether the engine shall be condensing or non-condensing, and if one large engine shall be used or whether the power shall be divided into several smaller units; each of these conditions will receive separate consideration.

KIND OF SERVICE.

53. Self-Selecting Service.—Evidently the service to which an engine shall be put is the first determining element in the selection of an engine. In many cases the kind of service at once determines the general type of machine, as, for instance, for hoisting, pumping, blowing, locomotive, and rolling service. There are many lines of service, however, in which the type is not self-selecting. The service may be divided into *continuous running with uniform load*, *continuous running with variable load*, *continuous running with uniform but increasing load due to growth*, and *intermittent running*.

54. Continuous Running With Uniform Load.—When the condition of continuous running with uniform load pertains, the first question arising is the cost of fuel. If this item of expenditure is great, the evident conclusion

is that the steam should be worked at as high a ratio of expansion as practicable, since the condition existing most readily permits economical working of steam machinery. Not only should the steam be worked expansively, but it should be done in a high-grade engine. Whether this be high-speed or slow-speed depends on the particular work the machine is called on to do. If the engine is direct-connected to an electric generator of not over 200 horsepower demand and the steam pressure is 125 pounds, while water is available for condensing purposes, maximum efficiency would be obtained by the use of a compound condensing high-speed engine having separate steam and exhaust valves and a governor controlling the admission valve only. Where economy is a desirable item, a high-speed engine having one valve for steam distribution is not a favorite. If the power demanded is large, the slow-speed high-grade engine should be the choice and the cost of fuel should dictate in a great measure the steam pressure, the ratio of expansion, and the refinements in all directions to the end of reducing all expenditures. If no water be available for condensing purposes, the compound non-condensing engine will prove a good investment; but to give good success, the compound non-condensing engine should have at least 140 pounds of steam pressure, and care must be taken that it is not too large for its work. It is better, as far as economy is concerned, to have this type of engine small rather than large for its work.

55. The prevailing practice for terminal pressures is 19 pounds absolute pressure in the United States, while 25 pounds absolute pressure is the practice of some good English builders. If expansion is carried below the atmospheric pressure, the low-pressure cylinder will prove a drag on the engine.

56. Continuous Running With Variable Load.—For continuous running with a variable load, the compound non-condensing engine should be avoided. For this service the compound condensing engine is most suitable, as it

works over a wide range of expansion without materially affecting its economical efficiency. The simple condensing engine is well suited for the purpose. If condensing water is not available and if the cost of fuel or demand for power is not sufficient to warrant the use of a cooling tower for condensing water, then the slow-speed non-condensing engine working with a steam pressure of 100 pounds should be the choice.

57. Continuous Running With Uniform But Increasing Load.—For continuous running with a uniform load, which is expected to be increased, however, through extension of the business, we turn naturally to the simple non-condensing engine of high or low grade, depending on the cost of fuel, and arrange to make it into a duplex engine, a condensing engine, or a compound condensing engine, if water is available, as demands are made for increase of power. It would be questionable economy to provide for converting this machine into a compound non-condensing engine on account of the high pressure required to successfully operate this type of machine. In the absence of condensing water, the increased power could be most easily and inexpensively provided by making the engine a duplex.

58. Intermittent Running.—For intermittent running there is much dispute among engineers as to the best type of engine. Here also the cost of fuel enters as a determining element of considerable weight. One of the most familiar examples of intermittent running is the hoisting engine. In the coal fields we find the simplest types of engines with no pretence at economy, while in the Northwestern copper-mining district we find the most elaborate triple-expansion hoisting engines working with 185 pounds steam pressure. Compound condensing hoisting engines are very common in the Northwestern iron-mining district and in the South African gold-mining industry. In both of these localities the cost of fuel is a heavy item; in South Africa it is not only expensive but poor in quality. Condensing water is also very scarce and the prevailing practice

is to make the engines compound condensing, using cooling towers to extract the heat from the water, and to use the same water continuously. The contention in respect to high-grade multiple-expansion engines for intermittent work is that if they are more economical in continuous service, the same or nearly the same comparative margin in their favor will result in intermittent work, and it must be conceded that there is reason in the contention.

59. The problem of choosing an engine for certain classes of intermittent work demands the study of local conditions, in which the cost of fuel is probably the most important determining condition. If the cost of fuel is high, say $5 per ton, and the power demanded is large, and the duration of work between stops is 2 minutes, the high-grade multiple-expansion engine should be a paying investment even though the extra expense of providing cooling towers for condensing water be added. If the power required is small, the high-grade engine is very seldom or never used, even though the cost of fuel be large. For reversing rolling-mill service, the simplest and strongest type of engine is used, the high-grade or multiple-expansion engine seldom being used.

60. There is another class of service that might properly be mentioned under the head of intermittent-running engines. This exists in industries that, from their nature, can operate only during a season or part of the year, such as the beet, cane sugar, certain classes of wood fiber, and other industries dependent on season and soil to produce their raw material. There is usually considerable refuse in these industries, having more or less value as a combustible, which, if it cannot be converted into a more valuable byproduct, is generally used as steam-generating fuel. The amount of refuse will quite often determine the type of engine to be used in running the plant. It is usually in excess (except in the beet-sugar industry, where the refuse is a marketable byproduct), and then the simplest, strongest, and cheapest constructed type of engine is chosen.

61. Relay engines can be classified to some extent as intermittent-running engines; but as their term of service is often of considerable duration, especially when they supplement water-power, with the further condition that many water-powers are decreasing in force year by year, the high-grade slow-running engine is generally chosen for this service for large powers and the better type of high-speed engine where smaller power or occasional assistance is required.

INFLUENCE OF LOCATION.

62. Fuel Cost.—The influence of location must be considered in conjunction with other influences, the principal one of these being the price to be paid for fuel; we mention this first because it is the largest and most important influence. It is evident that location is the all-determining influence of fuel cost, as an engine located in the coal-mining districts may be selected wholly with reference to low first cost, owing to the low fuel cost, while with an engine located in the iron- and copper-mining districts the first cost is a comparatively insignificant item if the power requirement is large, owing to the high price of fuel. This is not only true of the mining industries, but of all uses to which the steam engine can be put. In the New England States the cost of steam coal is about $3.50 per ton. Many of the installations are large, and here we find the best types of large economical steam engines. In many instances, the location is very remote from any source of supply or repair, which fact conduces to the selection of more simple machines, subject, however, to other conditions.

63. Cost of Transportation.—The cost of transportation of large engines working at a high ratio of expansion may influence the selection in favor of the smaller high-speed engine. A peculiar condition of transportation, due to location, is frequently met in the Western mountainous districts, where it is required to install an engine, usually for mining and metallurgical purposes, subject to the condition

that no piece shall weigh more than 500 pounds, the reason for this being that the parts must be transported on mule back through dangerous and difficult mountain passes. The simplest types of engines working without expansion and ingeniously divided into a number of parts determined by the size of the machine are usually chosen for these locations. Even here the item of fuel cost enters, which, with perhaps even the water supply, must also be transported in a similar manner as the engine parts; hence, high-grade engines working expansively have sometimes been chosen. It is almost needless to say that the first cost of such an engine is quite high, and that it taxes the skill of the designer and builder to the utmost to produce it.

64. Existing Conditions.—There are other conditions that follow as a natural result of location and which are not dependent on the cost of fuel or transportation, but which, to some extent, go far in determining the type of engine. Existing conditions at any given location may fix the type of machine, as steam pressure available, speed desired, and feasible method of coupling, and the availability of condensing feedwater. The demands of the particular service may require a special type of engine and frequently a machine of special and peculiar construction to meet the demands of peculiar local conditions, such as size and height of buildings.

INFLUENCE OF FIRST COST.

65. The item of first cost is probably the first to enter the deliberations of the constructing engineer and purchaser and throughout the determination stands out as a condition against which all other elements of the determination are weighed. Whether it be given first place or made second to the cost of fuel depends in a great measure on the amount of power demanded in proportion to the quantity of finished product. When the outlay for fuel is small compared with other running expenses, due to a small demand for power in the particular industry, low first cost may prevail; but, on

the other hand, if the power requirements are large, even though the price of fuel is moderate, the saving in fuel will soon overrun the interest and extra depreciation charges against the engine high in first cost, but economical in the use of steam. Where a large amount of power is required, even though the cost of fuel is low, if no other conditions enter the problem, such as remote location from any source of supply or facilities for repair or if the labor obtainable is untrustworthy or incompetent to work an efficient installation economically, it will generally be a wise investment to install the high-grade slow-running engine, taking advantage of water for condensation if possible and compounding if available steam pressures are not too low for good results.

INFLUENCE OF FUEL COST.

66. Introduction.—As was pointed out in Art. **65**, the selection of an engine is to a great extent determined either by the first cost or the cost of fuel. It must be understood that for special services and extraordinary locations and conditions, special engines must be designed to meet the condition, and to a great extent regardless of first cost or the cost of fuel. These are special cases and are almost self-solving.

67. Cheap Fuel.—When fuel is cheap and, as is sometimes the case, must be burned to dispose of it, an engine low in first cost will be the natural choice. This feature should be combined with simplicity, and the engine should be of such design as to require as little attention as possible, since where cost of fuel is of little or no consequence, the whole steam plant is liable to be neglected or left to care for itself, particularly the engine. In this case, one of the single-acting vertical enclosed-crank type of engine or some similar simple engine would be the natural selection.

68. Dear Fuel.—When the cost of fuel is high, recourse must usually be taken to every known means for saving fuel.

The extent of elaboration in that direction depends on the price of fuel. The various means and devices used at the engine to secure small consumption of fuel may be briefly mentioned, as high steam pressure and multiple expansion, steam containing 70° superheat, and steam-jacketing. It may be mentioned here that superheated steam and steam-jacketing are agents working in the same direction—namely, the reduction of initial cylinder condensation, and where one is used the other is superfluous; thus, to steam-jacket a high-pressure cylinder receiving steam of, say, 50° F., superheat would be a non-paying investment. A means of securing high economy in compound-engine performance is to provide an efficient reheating receiver.

Advocates of reheating receivers claim as high as 10 per cent. gain by their use. Low-pressure cylinders are frequently steam-jacketed, the pressure in the jackets being reduced to about one-half the boiler pressure. Many builders will not use jackets on low-pressure cylinders; the best practice, however, favors their use. Serrating or corrugating the outside of cylinder liners and jacket spaces of the heads to make them more active is sometimes practiced, as is also the polishing bright of surfaces exposed to the incoming steam. A thorough system of circulation and drainage of all jackets, as well as means for freeing them of accumulated air, are essential to high economy. Small units should be avoided, if possible, combining them into as few large units as practicable, since large engines are generally more economical than small engines. All cylinders, pipes, reheaters, etc. should be covered with a non-conducting covering.

69. If condensing water is available, it should be used and a vacuum of not less than 3 inches below the indication of the barometer should be obtained. If condensing water is not available, a cooling tower may be used, remembering that it is not possible to obtain a vacuum with cooling towers much better than 6 inches below the indication of the barometer. A primary heater may be used between the low-pressure cylinder and the condenser through which the feedwater

is pumped, thus extracting as much heat as possible from the steam. A steam separator should be used at the throttle valves, returning entrained water to the boiler. All other drains should be trapped to an automatic receiver pump to be returned to the boilers. Superheated steam, if heated to an effective stage, say 70° F. superheat, requires poppet valves on the high-pressure cylinder on account of the difficulty experienced in oiling and keeping any kind of slide valves tight. This type of valve is somewhat expensive compared with other types of valves. The superheat is so reduced by the time the steam reaches the low-pressure cylinder of a compound engine that Corliss or gridiron valves may be used on this cylinder. The degree of refinement to which it will be expedient for the engineer to go must be determined by the cost of fuel and the amount that the steam engine demands as against the interest and cost of repairs due to this extra outlay to secure the smaller outlay for fuel.

70. The growing practice in steam engineering is along the lines of greater economy of fuel, and the compound engine is fast finding its way in the coal-mining districts, even where the fuel is culm, which is delivered at a cost not much exceeding 50 cents per ton. Assuming that engineers in coal-mining districts have found it a paying investment to introduce high-grade engines for their purposes, there should be little question as to the advisability of elaboration in districts where the cost of fuel is not uncommonly eight times higher, provided no other militating influences present themselves.

INFLUENCE OF STEAM PRESSURE.

71. The steam pressure available may be a controlling factor where an engine is to be selected to replace an old or overloaded one. The steam pressure available is generally the element that determines whether an engine shall be a single-cylinder or multiple-expansion. Generally the steam pressure should be 100 pounds gauge pressure for effective

compounding with a condenser, while 135 pounds should be available for compound non-condensing engines and 160 pounds for triple-expansion condensing engines. It is a matter of fact that the compound condensing engine has such an economical range over wide variations of steam pressure that it can be proportioned to secure results approaching so nearly the triple-expansion engine that the additional outlay for a triple-expansion engine is questionable, except for special and otherwise favorable service, such as high-duty municipal pumping engines, where three cylinders, each actuating a separate plunger, conduce not only to extreme economy of steam, but to a steady flow of delivery water, thus avoiding shocks on both pumping machinery and delivery mains.

INFLUENCE OF DURATION OF SERVICE.

72. By duration of service is often meant the life of the engine for useful work. It frequently happens that machinery is installed for the purpose of developing a doubtful industry or on speculation, when the measure of the doubt will be the controlling influence in determining the degree of efficiency and first cost of the engine. Engines and machinery are sometimes sent to distant localities or those difficult of access to perform a service, and the expenses of transportation are such as not to warrant their return, for it must be borne in mind that when an engine has been used sufficiently to be called second hand, its selling price is reduced below its former value. In such cases, engines of low first cost, but strong, simple, and well built, are usually selected. It is manifestly important in such cases that accident and costly delay by breakage be provided against.

INFLUENCE OF FACILITIES FOR REPAIRS.

73. While this influence is usually not a strong or active one, it nevertheless must be considered and provided for either in the selection of the engine or provisions for keeping

it in good working order. This latter may be accomplished by providing the necessary skilled labor, tools, and supplies or by providing spare parts to replace such pieces as have been found by experience most likely to break.

Specifications for engines for foreign shipment frequently include the following parts: 1 pair of connecting-rod brasses for each end of rod; 1 crosshead shoe; 1 piston and rod complete; 1 complete set main and outboard bearing boxes; 1 eccentric strap; 1 complete releasing gear with dashpot, if of the releasing-gear type of engine; 1 steam valve and 1 exhaust valve for each engine or each side of a compound engine. If a condensing engine, the following additional parts are usually specified: 1 air-pump bucket and rod; 1 air-pump delivery deck with valve and guards complete; 1 set of India-rubber valves; duplicate sets of metallic packing to be furnished and all stuffingboxes designed for the use of fibrous packing and suitable glands to be provided.

74. While the influence of facilities for repairs may in many instances of small and even moderate-size plants dictate the simplest and strongest types of steam engines, it does not, in fact, obtain in the case of large installations, where the cost of fuel is high or even moderate. The difficulty can be met by providing either spare parts, facilities for repairs, or relay engines, and in many of the South African plants all three expedients have been found desirable. Even with the simplest and strongest types of engines, if the facilities for repair are not at hand, carrying spare parts is advisable, as accidents and defects are liable with the most carefully constructed engines.

INFLUENCE OF KIND OF LABOR OBTAINABLE.

75. This influence is one that to a great extent is controllable, but the possibility of being compelled to trust an expensive steam plant in unskilled hands even for short periods, and possibly for long ones, may have some weight

in determining the type of engine. High-grade economical engines generally require superior intelligence to maintain them in that condition where the item of extra first cost may be assured as a constant and continuing profitable investment. The condition of the kind of labor obtainable may, when other conditions operate against the selection of high-grade engines, carry the choice to the simplest and strongest types of machine, but except in extreme cases it is not of sufficient weight itself to determine a selection.

INFLUENCE OF EXISTING CONDITIONS AND PROBABLE EXTENSIONS.

76. This influence has to a considerable extent been dealt with in former articles, but it will not be out of place to retrace the subject, as it is important, and a little forethought in this direction may save a considerable future outlay with less satisfactory results. Additional engine power may be obtained in many ways. One of the simplest and least expensive practices is the addition of a condenser where condensing water is available, the resulting increase of power ranging from 18 to 25 per cent., depending on the type of engine and degree of vacuum obtained. A good condenser should add from 11½ to 12 pounds to the mean effective pressure. A jet condenser at average temperatures will require about 28 times as much injection or condensing water as there is steam to be condensed; that is, 28 pounds of water at 70° F. will be required to condense 1 pound of steam at 19 pounds absolute pressure. The duplex engine is an excellent and efficient means of increasing power, as is also compounding by adding a low-pressure cylinder if condensing is practicable, even at the expense of a cooling tower. Another means of increasing the power and securing economy where provisions for the duplex engine have not been made is by installing the low-pressure side of a compound by means of an entirely separate engine and running them disconnected. Such engines act with sufficient precision for all practical purposes.

CONDENSING OR NON-CONDENSING ENGINES.

77. The question of whether to run condensing usually depends for its answer on the natural supply of cooling water available, and frequently the supply of cooling water determines the location of the steam plant. This is particularly true of large installations, which, if in large cities, are invariably on the water front. There is a decided gain by the use of a condenser, not only in fuel, but in first cost, as a condensing engine may be made, on the average, 20 per cent. smaller and almost 20 per cent. cheaper than one not provided with a condenser. The cost of air pump and condenser, if directly connected, the air pump being driven by the main engine, should be about 10 per cent. of the cost of the engine for average sizes, and should require about 2 per cent. of the power of the engine to drive it. If an independent condenser is used, which for many reasons is the most desirable arrangement, it should cost about 15 per cent. of the cost of the engine, considering here slow-running high-grade engines. High-speed and inferior engines require condensers larger in proportion, owing to the larger amount of steam used. For large plants, where abundant natural water is not to be had, the cooling tower may be used to extract the heat from the injection water and to use it repeatedly. Sometimes a pond or large pans or tanks on the roof top are devised for the purpose of cooling injection water. Both of these plans of cooling water are slow and very large areas are required and, on account of atmospheric changes, are very uncertain.

78. A number of quite effective water-cooling devices are now being regularly manufactured in units as large as 10,000 horsepower. The general principles of all are the same; they consist of a round or rectangular tower so devised that the delivery water, which the air pump delivers to the top of the tower, in its descent is divided into the greatest possible number of sprays or films; an artificial current of air traverses the surface of the water, extracting the heat from it and rendering it sufficiently cool for service as injection water. These towers, as now constructed, do not require much floor

space, a 300-horsepower tower occupying a space of 8 feet × 12 feet. Self-cooling condensers have also been used to avoid steam plants becoming a nuisance in thickly populated city districts, by virtue of the suppression of all exhaust noises, although the law allows an industry to make as much noise as is reasonable and unavoidable in the pursuance of its processes. The cost of fuel usually dictates the policy in regard to cooling towers. If a natural water supply is available, it should be taken advantage of for economical reasons. The jet condenser is the favorite for all land purposes on account of its low first cost as compared with surface condensers.

ONE LARGE ENGINE OR SMALL UNITS.

79. This is a problem not only demanding large experience and most careful study of the details, but it also demands a broad and comprehensive investigation regarding future conditions. The conclusion to put in one large unit is too frequently jumped at. Generally the fewer units there are, the more economical will be the plant; but the condition of service goes far to determine the selection in this respect. In many industries departments require the use of power only a short time or at intermittent periods during the day, and frequently demand widely varying speeds for best effect; they also frequently require to be operated overtime or all night. Long-distance transmission is also involved. In such cases it will often be found that subdivided power will give the best results and, as a matter of fact, some large industries in the New England States formerly driven by one large single unit have adopted the scheme of subdivided power, dividing a single 1,400 horsepower unit into 40 smaller units of varying power. The high efficiency of electricity as a power-transmitting medium has done much to solve the problem of transmission to remote and difficult points and has also contributed to the existence of large single units; but these units should not be so large that they will be run underloaded, for an underloaded engine is about as poor an investment as can well be imagined.

ENGINE FOUNDATIONS.

80. Purpose of Engine Foundations.—One of the most important items in the installation of engines is to provide a suitable foundation, not only in order to rigidly support the machine, but also to absorb the jars and shocks due to its reciprocating motion, because if these are not absorbed, it will result in injury to the engine in question and also to adjacent property, such as other foundations, walls, and structures of any kind resting on the adjacent soil.

81. Supporting Power of Soils.—The foundation, besides having sufficient mass to absorb vibrations, should be spread out over sufficient area to prevent settling. Accepted figures for the supporting power of various soils range from 1 ton per square foot for soft clay to 5 tons per square foot for compact sand bottom, while 200 tons per square foot is given as the supporting power of hard rock in thick strata.

82. Depth of Foundations.—The depth of a foundation will vary with conditions; it should go out far enough below the surface to be free from the effects of frost or the influence of loads borne by adjacent grounds. It is rarely less than 4½ feet for small engines and rarely exceeds 22 feet for the largest engines. A horizontal slow-speed engine foundation for a 40-horsepower engine should be 7 feet deep; a 200-horsepower engine foundation should be 9 feet 6 inches deep; a 600-horsepower engine foundation should be 11 feet 6 inches deep.

Different types of engines require somewhat different designs for their foundations; experience has been and is the only teacher in this subject.

83. Absorption of Vibrations.—High-speed engines stop and start the reciprocating parts many times a minute, and hence set up severe vibrations, which must be absorbed by the foundations. In many engines of this type very careful counterbalancing is used to balance the reciprocating

parts in the horizontal direction; this, however, leaves the counterbalance unbalanced in the vertical direction. This balancing in horizontal engines tends to prevent the engine sliding lengthwise upon its foundation, while in the vertical engine the revolving counterbalance tends to slide the engine upon its foundation in a horizontal plane, and the foundation in either case must be of sufficient mass to absorb the vertical and horizontal forces, since engine framings and subbases rarely have sufficient mass to absorb vibrations. When engines are on the upper floors of a building, the scheme of suspending a very heavy mass underneath the floor, but rigidly bolted through to the engine base, and thus making virtually a foundation suspended in air, has proven effective in preventing all vibrations. Care must be exercised in placing engines upon solid rock that some elastic medium, as layers of wood and hair felt, is used between the machine and the rock to prevent vibration of the engine being transmitted to adjacent property.

84. Foundation for Vertical Cross-Compound Engine.—In a vertical two-crank high-speed engine having the cranks placed opposite each other, the vertical forces act to vibrate the machine in a vertical plane parallel to the crank-shaft. Such foundations should be designed with footings, as *a* and *b*, Fig. 1, relieving the center of the foundation and thus preventing any tendency to rock on a central bearing. Fig. 1 gives the general foundation dimensions for a 700-horsepower vertical cross-compound engine running at 100 revolutions per minute.

85. Comparison of Foundations for Vertical and Horizontal Engines.—Horizontal engines usually occupy so much space in a horizontal plane that the supporting power of the soil will be very much above the load if the foundation is made as small as possible—that is, if the bolt holes through the capstones are 6 inches from the center of hole to the edge of the stone for a 1-inch bolt and 10 inches for a 3-inch bolt. There should be 4 to 8 inches of masonry outside the capstone, and if the sides are carried down

straight, sufficient bearing area will be covered, except in cases of alluvial soil. Vertical engines, owing to the small

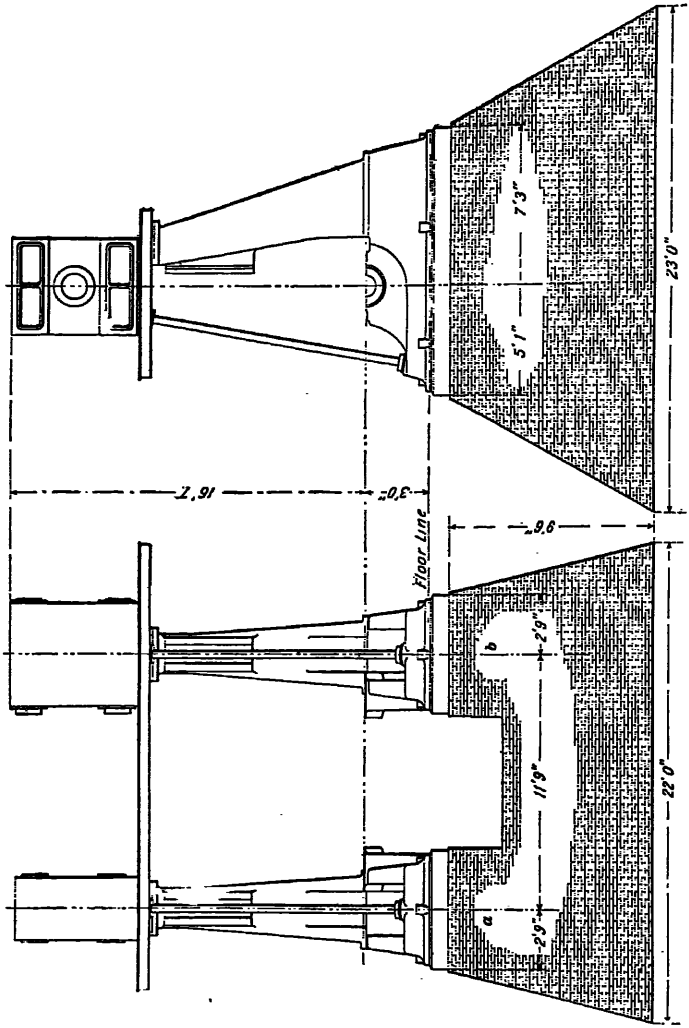


FIG. 1.

horizontal space required, should have deeper foundations than horizontal engines, and to secure sufficient bearing area,

the sides may be battered to any desired extent. Bearing surface for vertical engines should be carefully calculated with reference to the nature of the supporting soil, including, of course, the weight of the foundation itself as well as the engine that it supports.

86. Foundation Material.—The material of which a foundation is made depends very much on the location and the kind of material available. Brick is the most common material; dressed stone laid in cement mortar is sometimes used; concrete is growing in favor for engine foundations. When brick is used, it should be first quality hard brick laid in Portland cement mortar; lime mortars are not suitable for engine foundations on account of their tendency to disintegrate under vibration. Stone foundations should also be laid in cement mortar.

Foundations of concrete are coming more into use; they are constructed by first providing a level and suitable footing upon which a casing of timber, embracing the outlines of the foundation, is built. This is open at the top and bottom. The foundation bolts are suspended in pipes, old boiler tubes, or wood launders, leaving a space of at least 1 inch all around the bolt; successive layers of cement concrete are thrown in and well rammed until the desired height is reached.

87. Foundation Footings.—Foundation footings are in some cases required, most frequently on the water front, where it is necessary to go many feet deep to find a sufficiently hard bottom to support the load. This is accomplished most commonly by piling, which consists of driving long sticks or timbers, as *a*, *a*, Fig. 2, down to hard bottom, placing them 2½ to 4 feet apart from center to center. A timber grating *b* is fastened to the tops of the piles and a layer of concrete *d* is deposited. Planking, as *c*, *c*, is sometimes put on the framing, which distributes the pressure, but it is considered objectionable, as it prevents any connection between the superstructure and the concrete and

increases the liability of sliding. The space between the piles is frequently filled with rubble, clay, or concrete, and upon this footing the foundation proper is built. A properly driven pile, well supported against lateral flexure, may bear from one-eighth to one-tenth the crushing load, which

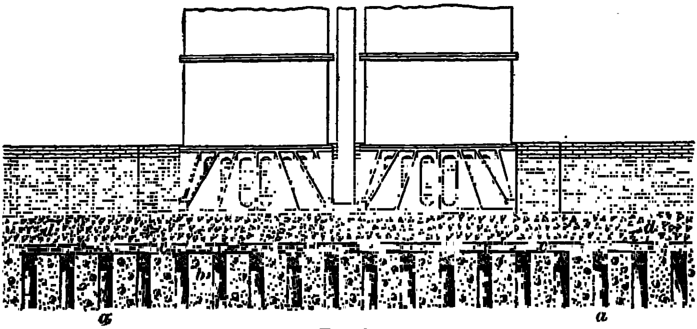


FIG. 2.

varies between 5,700 and 8,500 pounds per square inch. A pile 7 inches in diameter will bear about 12 tons. A pile can support a load of 25 tons when it refuses to move more than $\frac{3}{8}$ inch under thirty blows of a monkey weighing 1,200 pounds and falling 4 feet.

88. When rock is struck at a high level, special footings to prevent vibration must be constructed. An underlying stratum of timber or rubble or of both has been tried with questionable success; a layer of 2 or 3 feet of sand constrained laterally by a casing to prevent displacement has proved quite effective. The sand is also filled in around the sides of the foundation block. A heavy layer of asphalt is also effective in breaking engine vibrations before they reach the transmitting rock upon which the foundation is built.

89. Capstones.—Brick and dressed stone foundations usually require capstones to make a good job. These are usually granite and vary in thickness from 8 to 24 inches. Concrete foundations usually require no capstones. Instead of capstones, cast-iron sole plates are sometimes used. They

are usually thin, about $\frac{7}{8}$ or 1 inch thick, with an upturned ledge around the top to keep oil from the foundation and sufficient ribs below to give stiffness to the plate, and are provided with raised planed facings to match the engine parts. They are not more expensive than good capstones and are a superior job. Every precaution should be taken to keep oil from reaching the foundations, as it will dissolve the cement.

90. Many erecting engineers make a practice of setting the capstone for the outboard bearing from $\frac{1}{4}$ to $\frac{1}{2}$ inch lower than the actual figures called for and shim up the sole plate with wrought-iron strips or plates, the contention being that if the stone is set low the bearing can be shimmed up, but if a little too high it is a difficult matter to do anything with it but to chip off the top or take it up and reset it.

91. Foundation Bolts and Washers.—The bolts are always made of wrought iron, and should be of good quality, as Burden's best-best, Catasauqua, or some equivalent brand. Foundation bolts are usually made in length nearly the full depth of the foundation, and in important work they are made upset, that is, the threaded portion is made enough larger in diameter that the bottom of the thread is still a little larger than the body of the bolt. By this means the stretch due to the pull on the bolt is distributed over the long body and not only over the threads, as would be the case if the thread were cut on a rod of uniform diameter. Foundation bolts vary in diameter from $\frac{7}{8}$ inch in small engines to 4 inches in the largest types of land engines.

92. Foundation washers are commonly made of cast iron, but for small engines wrought-iron plates from $\frac{3}{8}$ to $\frac{3}{4}$ inch thick are used. In many locations it is not possible to provide pockets for access to the foundation washers; in such cases it is the best practice to provide cast-iron

washers *a*, Fig. 3. With this style of foundation washer it is possible to adjust the bolts to any desired height, or even to remove them and replace them at will. Some builders make a practice of tapping the top of the foundation bolt for an eyebolt to facilitate lifting or lowering the bolt in place.

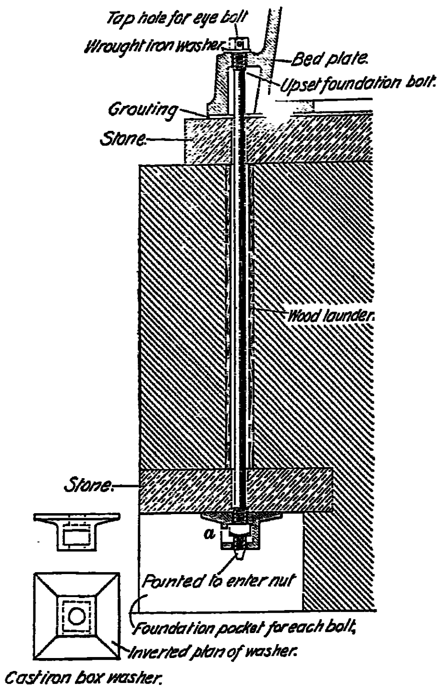


FIG. 3.

93. In large work it is quite important to have the foundation bolts removable, for if they extend through high framing or high bosses, it is necessary to lift the castings over the top of the bolts, which adds much to the cost of erection.

94. With many of the mining companies it is the custom to build foundations with pockets according to the drawing, but omitting the holes for the bolts entirely. When the capstones are leveled and grouted, they lay off the holes from the castings and drill holes for the bolts with a diamond drill. Sometimes the engine is erected, lined up, and grouted, and then the holes for the foundation bolts are drilled in place. With this arrangement it is essential in the design of the machine to see that holes for foundation bolts are not covered by any part or projection of the machine, or if this cannot be done, to provide a hole to pass the foundation bolt through.

95. Foundation Templets.—Foundation templets are used for the purpose of accurately locating the bolts, bringing

them to the proper height, and holding them in position while the masonry is being built. They are constructed of wood with blocks of varying height to suit the height of the engine bosses. The center line of the engine is carefully marked on the templet with correct relation to the bolts, and at right angles to it is marked the center line of the crank-shaft. Suitable marks and dowels to facilitate putting it together, if of such dimensions that it is necessary to ship it in sections, are also provided. The foundation bolts should not be allowed to hang on the templet, but rest on stone or bricks. The bolts, if they have no adjustment, should be set originally from 1 to 2 inches higher than required, as the gradually increasing weight of the foundation will sink the soil upon which it was started, and hence the bolts may not project through the engine casting unless this precaution be taken. Bolts when used in pockets or box washers should be pointed to facilitate entering the nut. Templets for out-board bearings or compound or triple-expansion engines are usually not connected, the relative locating being done from the foundation drawing. This work is usually done by the engine contractor, who sends only skilled men for this duty, but if not done by the engine contractor, it should be checked and approved by him at a sufficient time before the work of erection commences to have all defects, real or alleged, made good satisfactory to both parties.

96. Supporting the Templet.—The supporting of tne templet must be left to the ingenuity of the erecting engineer; generally it should be supported outside of the foundation, but there is no real objection to building the supporting posts into the mass and sawing them off when the foundation has reached about 18 inches from the top.

97. Setting the Templet.—Setting the templet is a simple matter, but it must be carefully and exactly done, especially if the engine is to drive a shaft by belt or gearing. The first and most important thing to do is to have the templet exact; particularly the crank-shaft center line must be square with the center line of the engine. This can be

tested by measuring off from the intersection of the two lines 6 feet on the shaft center line and 8 feet on the engine center line and adjusting the lines with their intersection as an axis until the hypotenuse of the triangle measures exactly 10 feet. Then having given the templet the correct relative position and the correct levels, the only remaining thing to do is to set the center line of the crank-shaft parallel to the line shaft or its established line. This can be done by plumbing down from or to the line shaft and measuring at both extremities of the crank-shaft center line from that line to the plumb-lines. Some mechanics establish the center lines of the engine and the crank-shaft by stretching the lines considerably above the templet height and set the templet from these lines by plumbing down. It is not a very easy matter to stretch two lines in the air at exactly right angles, and hence it is a better plan to have the lines on the templet and exactly right, and then to work to the crank-shaft line as the most important one. Having properly set the templet, the bolts are passed down through the holes and the washers and nuts put into place. Each bolt must rest on a large stone slab. Old pipe or wooden boxes should be put around the bolts to allow considerable lateral adjustment of the bolts.

98. Placing the Engine.—When the foundation is built up to within 2 feet of the top, the templet is removed and the top of the foundation built and carefully leveled by means of sensitive levels and straightedges. If the engine is large and the wheel is in halves, one-half the wheel should be placed in the wheel pit first; then the framing, outboard bearing, and shaft may be placed and finally the cylinders and valve gears. The engine is leveled in a plane parallel with the center line of the shaft and cylinder by means of sensitive spirit levels and all parts resting on the foundation are wedged and shimmed up. The bolts are tightened down moderately and the space between the bed-plate and foundation, which will vary from $\frac{1}{4}$ to $\frac{5}{8}$ inch, is filled with grouting.

99. Grouting.—The grouting may be made of iron borings mixed with cement, sal ammoniac, sulphur, and water in about the following proportions: 2 parts of sal ammoniac, 1 part of sulphur, 5 parts of cement, and 40 parts of iron borings mixed with enough water to make a heavy paste. Sometimes melted sulphur alone is used, but one of the very best groutings and the most easily applied is pure Portland cement. The rust joint must be driven in, while the sulphur and cement will flow in, suitable dams being constructed to constrain it to its proper place. Bolt holes should be filled with liquid grouting. Some builders who use box bedplates fill the entire bedplate with concrete to give it solidity and to reduce the tendency to magnify knocks into pounds.

100. Setting the Outboard Bearing.—The setting and alining of the outboard bearing should be carried along with the progress of the other work, but its final adjustment is important and should be done last, by the aid of lines representing the center lines of the cylinder and crankshaft. All outboard bearings should be provided with a sole plate and means for lateral adjustment either by screws or wedges, while some are provided with means for vertical adjustment.

PUMPS
(PART 1A)

STEAM PUMPS

DIRECT-ACTING STEAM PUMPS

INTRODUCTION

1. A **pump** may be defined as a device that will force a fluid against pressure. Most pumps are intended for use with liquids.

In water pumps operated by steam, the combination of parts through which the motion is imparted to the water is called either the **water end** or the **pump end,** while the combination of parts through which the force of the steam is applied is called the **steam end.**

A **reciprocating pump** is one in which the part impelling the liquid has a *to-and-fro motion,* while a **rotary pump** is one in which the impelling part has a *rotary motion.*

2. Reciprocating pumps are divided into two general classes according to the way in which the piston or plunger acts on the liquid. If the piston or the plunger delivers on every second stroke, the pump belongs to the *single-acting* class; if the delivery occurs on each stroke, the pump is of the *double-acting* class. If the pump uses a piston, it is called a **piston pump;** if a plunger is employed, the pump is said to be a **plunger pump.** If the piston or plunger moves horizontally, the pump is called a **horizontal pump,** while in case the piston or plunger moves vertically, the pump is called a **vertical pump.**

Steam pumps may be subdivided into *direct-acting* and *flywheel steam pumps.* **Direct-acting pumps** have no rotating parts, and the steam and water pistons are connected to the same piston rod, or rods, and move back and forth together in their cylinders. **Flywheel steam pumps** are used for the same purpose as the direct-acting type. The purpose of the flywheel is to allow an early cut-off, so as to use the steam expansively.

It is customary to refer to large pumps of the flywheel pattern as **pumping engines.** Direct-acting steam pumps are commonly classed as *single* and *duplex pumps.* The former has one water cylinder and may be either single- or double-acting. A duplex pump consists of two single pumps placed side by side and so constructed that the steam valve of one is driven from the piston rod of the other.

3. Pumps are frequently so located that the water must flow into the pump cylinder by atmospheric pressure on the surface of the water external to the suction pipe; that is, by the action of the pump a partial vacuum is produced in the pump chamber. If the end of the suction pipe, which is the pipe connecting the pump chamber with the water, is submerged, the excess of pressure on the surface of the water outside of the suction pipe will cause the water to rise in the suction pipe until the pressure due to the weight of the column equals the pressure of the atmosphere.

The pressure of the atmosphere is constantly changing. For practical purposes the pressure at sea level is taken as 30 inches of mercury, or 14.7 pounds pressure per square inch. Since a pressure of 1 pound per square inch is equal to that exerted by a column of water 2.309 feet high, the theoretical height that water can be raised by a perfect vacuum at sea level will be $14.7 \times 2.309 = 33.94$ feet. Since the atmospheric pressure becomes less as the altitude increases, it follows that the greater the altitude, the less the theoretical and practical lift by atmospheric pressure will be. To find the theoretical height in feet to which water can be lifted at any altitude, multiply the barometric reading in inches by 1.133.

For water holding foreign substances in suspension, or for other liquids, the theoretical lift can be found by dividing the theoretical height to which water can be lifted at the existing atmospheric pressure, as shown by the barometer, by the specific gravity of the liquid.

Since a perfect vacuum cannot be obtained on account of mechanical imperfections, air contained in the water, and the vapor of the water itself, the actual height to which it can be lifted is only about .82 of the theoretical height, which ratio is good only for pure water.

4. The lifting of hot water is a difficult problem, and experience has shown that it is not possible to lift water at all with a pump when the temperature of the water exceeds about 180° F. If hot water must be handled by the pump, this should be so located that the water can flow to it by gravity.

5. The limit of height to which any liquid can be forced by a pump is not affected by the atmospheric pressure and is only limited by the power available for forcing the liquid and the strength of the pump and the pipe connections.

6. The direct-acting steam pump is the most widely used form of steam pump. In pumps of this construction the moving parts have no weight greater than that required to produce sufficient strength in such parts for the work they are expected to perform; as there is, consequently, no opportunity to store up power in one part of the stroke to be given out at another, it is impossible to cut off steam in the cylinder during any part of the stroke. The uniform and steady action of the direct-acting steam pump is dependent alone on the use of a steady, uniform pressure of steam throughout the entire stroke of the piston against a steady, uniform resistance of water pressure in the pump. The difference between the force exerted in the steam cylinders over the resistance in the pump governs the rate of speed at which the piston or plunger of the pump will move. The length of the stroke of the steam piston within these pumps is limited and controlled by the admission, release, and compression of the steam in the cylinder.

SINGLE DIRECT-ACTING STEAM PUMPS

7. In Fig. 1 is shown a perspective view of a **Knowles steam pump,** built by what is now the Blake-Knowles Steam Pump Works, East Cambridge, Massachusetts, designed for boiler feeding, and fitted with a hand-power attachment that permits the pump to be used when no steam is available. The pump, which is a single one, consists of the steam cylinder *a* placed in line with the water cylinder *b*; the steam piston and water piston are both attached to the piston rod *c*. The suction

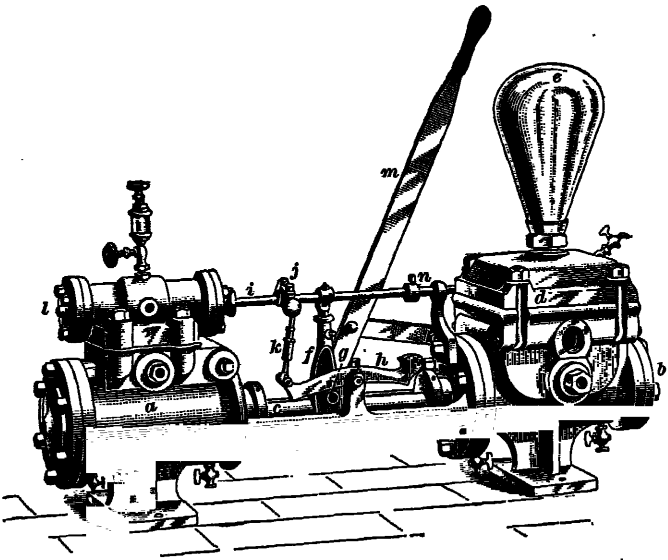


FIG. 1

and delivery valves are contained in the valve chest *d*, on top of which is placed the air chamber *e*, which promotes a steady delivery of water. The piston rod *c* carries an arm *f* that is provided with a stud carrying the roller *g*, which moves back and forth with the motion of the piston rod under the curved rocker bar *h* and lifts it near the ends of the piston stroke, thereby giving a slight rotary motion to the valve stem *i* by means of the crank-arm *j* and link *k*. This rotary motion of the valve stem is transmitted to a valve inside the steam chest *l*,

which valve, in turn, works the main valve in the manner described farther on.

For hand operation, the lever *m*, which has a fork on its lower end, is dropped over a stud on *f* and worked back and forth, thereby moving the piston rod *c*, and hence the water piston and incidentally the steam piston. This hand lever is

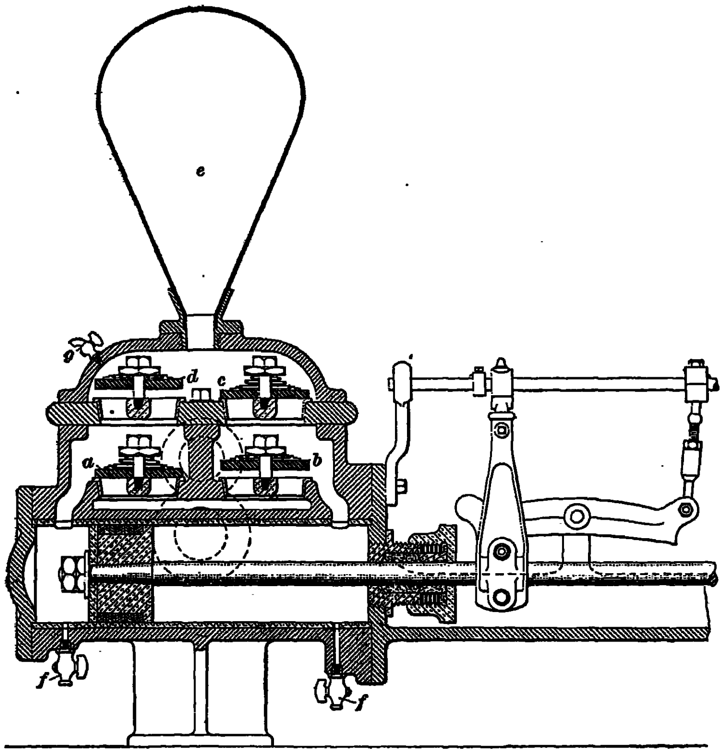


FIG. 2

fitted only to the smaller sizes of the Knowles single direct-acting pumps.

8. The water end of the Knowles pump, which is double-acting and of the piston type, is shown in Fig. 2. There are two suction valves *a, b*, and two delivery valves *c, d*. The air chamber *e* is open at the bottom, as shown. Drain cocks *f, f*

are used to drain the pump; a cock *g*, called an *air cock*, is screwed into the cover of the valve chest and when open permits a free escape of air from the space above the delivery valves; this cock is closed as soon as water issues from it.

With the piston moving to the left, the suction valve *b* and delivery valve *d* are open and the valves *a* and *c* are closed; with the piston moving to the right, the valves *a* and *c* are open and the valves *b* and *d* are closed.

The water passing the delivery valves partly fills the air chamber *e*, compressing the air contained therein; the elastic cushion thus formed keeps the water in the delivery pipe in motion when the piston is at the ends of its stroke.

9. A sectional perspective illustration of the steam end of the Knowles pump is shown in Fig. 3. The steam piston *a* is almost at the end of its stroke and the auxiliary valve *b*, sometimes called the *chest piston*, is ready to reverse. When the roller on the piston rod engages the rocker bar *c*, it raises the link *d* and thus rotates the chest piston, thereby bringing the port *e* opposite the steam inlet *f* and the port *g* opposite the exhaust port *h*. The pressure of the live steam on the left of *b* now moves it to the right, carrying with it the main valve *i*, which shuts off the live steam from the left side of the piston *a* and opens the main port *j* to the exhaust. The main valve also closes the auxiliary steam port *k* leading to the extreme left of the cylinder, opening at the same time the main port *l* and auxiliary port *m* to the live steam. This, entering the cylinder through the auxiliary port *m*, starts the piston gradually and smoothly on the return stroke; the piston in its onward motion uncovers the main steam port *l* and then receives the full steam pressure.

When the piston nears the end of the stroke, it closes the main steam port at that end and thus traps exhaust steam in the space between the piston and cylinder head, as *n*, which cushions the piston by arresting its motion gradually, and prevents its striking the cylinder head. The main steam ports *j*, *l* serve, alternately, for live and exhaust steam, as in an engine. When the main valve *i* is in the position shown, the main port *j*

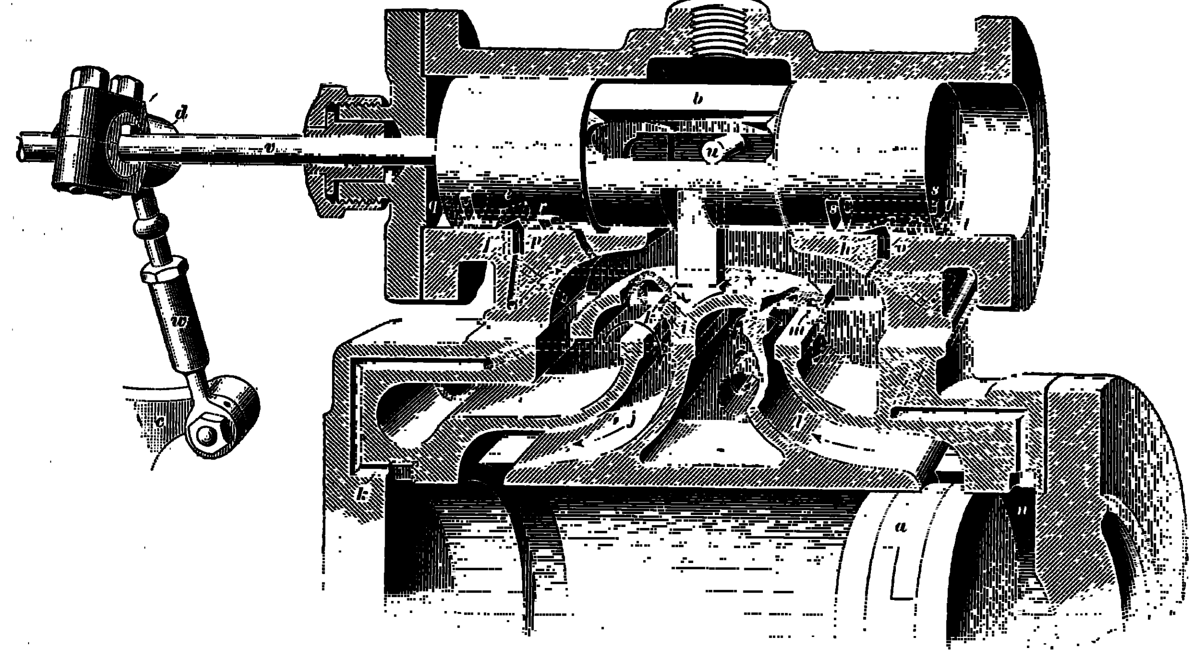


FIG. 3

is used for live steam, which enters as shown by the arrow; the port *l*, up to the time of its opening into the steam cylinder being closed by the piston *a*, served as an exhaust passage, the steam escaping as shown by the arrows. When the piston *a* has almost reached the end of its stroke to the left end of the cylinder, the roller carried by the standard fastened to the piston rod engages the rocker bar *c* and pulls *d* downwards, thus rotating the chest piston *b* in a direction opposite to that of the previous stroke. This rotation brings the port *g* opposite the steam port *o* and the port *e* opposite the exhaust port *p*; the live steam now entering the right end of the auxiliary cylinder forces the chest piston *b* to the left into the position shown in the illustration.

10. Under ordinary conditions, when everything is properly adjusted, the ports *q, r, s, t* in the chest piston *b*, Fig. 3, have no part in the operation of the pump. They are merely safety ports designed to cushion the chest piston with live steam should it be carried beyond its normal extreme positions. Ordinarily, the chest piston is cushioned by exhaust steam; should it be carried too far to the left, the port *r* will register with the live-steam port *f*, thereby admitting live steam to the left end of the chest piston at the same time that the port *t* will register with the exhaust port *h*. If the chest piston is carried too far to the right, the ports *q* and *p*, and *s* and *o*, will register. A bolt *u* engaging a slot in the chest piston limits its rotation.

11. The arm *f*, Fig. 1, that carries the roller engaging the rocker bar *h* (*c*, Fig. 3) is extended upwards and encircles the valve stem *i*, Fig. 1 (*v*, Fig. 3), and strikes either the rocker-arm *j*, Fig. 1 (*d*, Fig. 3), or the collar *n*, Fig. 1, and thereby moves the chest piston mechanically in case it should fail, for some reason, to move automatically. The arm *f*, Fig. 1, however, does not touch either the rocker-arm or the collar on the valve stem during the ordinary operation of the pump; it will also touch the rocker-arm or collar and thereby move the chest piston when the main piston for any reason exceeds its usual length of stroke.

The stroke of the pump is equalized by changing the length of the link *k*, Fig. 1 (*w*, Fig. 3), until the roller engages the rocker-arm at points equidistant from the rocker-arm fulcrum.

12. In Fig. 4 is shown in section the steam end of a **Cameron** single direct-acting pump, the valve mechanism of which is entirely enclosed. This pump is built by the A. S. Cameron Steam Pump Works, New York City, New York. The steam cylinder *a* has small cylinders *b, b′* at the upper edge

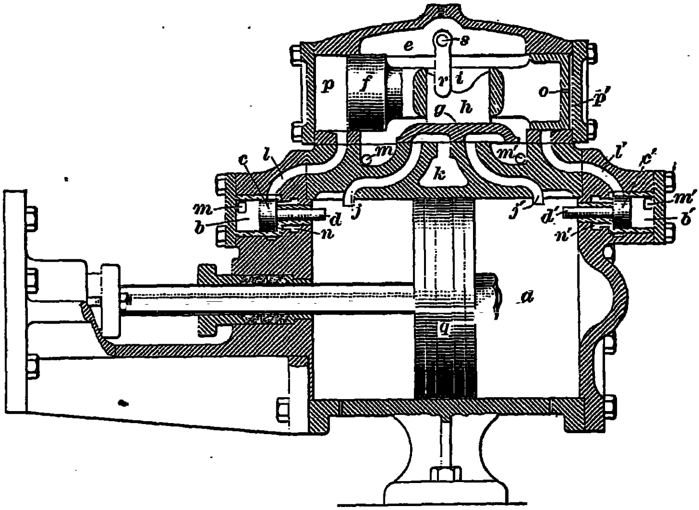


FIG. 4

of each head. These cylinders contain pistons *c, c′*, called *reversing pistons*, with piston rods *d, d* projecting a short distance into the pump cylinder. The steam chest *e* contains a hollow, double-ended plunger *f*, with the ends fitting a cylindrical cavity at each end of the chest. The main slide valve *g*, of ⊓ form and with a rectangular projection *h* on its back, extends into a slot *i* in the central portion of the double-ended plunger *f*. Ports *j, j′* connect the cylinder with the steam chest or the exhaust port *k*, according to the position of the slide valve *g*. The ports *l, l′*, and *m, m′* connect the steam chest *e*

with the small auxiliary cylinders *b, b'*. The exhaust port *k* is also connected to the cavities *n, n'* at the inner ends of the cylinders *b, b'*.

13. When steam is admitted to the steam chest *e*, Fig. 4, it passes through small holes *o* in the ends of the double-ended plunger *f*, filling the spaces *p, p'*, the ports *m, m'*, and the spaces *b, b'* behind the reversing pistons *c, c'*. When the plunger *f* and the slide valve *g* are in their right-hand positions, as shown in the illustration, steam will be admitted directly from the steam chest *e* to the cylinder *a* through the port *j'*, and the piston *q* will move to the left. The steam at the same time is exhausted from the left end of the cylinder *a* through the port *j* and the exhaust port *k*. The piston *q*, when near the end of its stroke, strikes the projecting end of the rod *d* and moves the small piston *c* to the left, thus exhausting the steam from the port *l* and space *p* at the left end of the plunger *f* through a small exhaust passage leading from the space *n* to the exhaust port *k*. The pressure in the space *p* thus being lowered, the much higher pressure of the steam in the space *p'* at the right-hand end of the plunger *f* forces the plunger, with the valve *g*, to the left. In this way steam is admitted to the left-hand end of the cylinder *a* and exhausted from the right-hand end, so that the motion of the steam piston *q* is reversed. The small pistons *c, c'* are held at their inner positions by live-steam pressure through the ports *m, m'*. Thus, steam is prevented from exhausting from the ports *l, l'*, which are alternately uncovered for only a short time when the steam piston is at the end of its stroke. As steam cannot escape from the ports, it is not necessary to pack the ends of the plunger *f*, and there is little frictional resistance to its movement. This plunger moves with great rapidity, and, by shifting the valve *g*, admits steam in front of the piston in time to overcome its momentum by cushioning. The piston *f* may be moved by hand by means of an arm *r* that engages the piston and is attached to a shaft *s* passing through a stuffingbox in the side of the steam chest. By this means, the slide valve may be moved so as to reverse the pump when it fails to do so automatically, as, for instance,

when chips or dirt clog the valve, or in case of neglect, when the piston valve and seats may be rusted.

14. The **Marsh** single direct-acting steam pump, built by the American Steam Pump Company, Battle Creek, Michigan, employs a steam-thrown valve; the steam end of this make of pump is shown in section in Fig. 5. The steam piston *a* is made in two parts that are so arranged as to provide an annular space

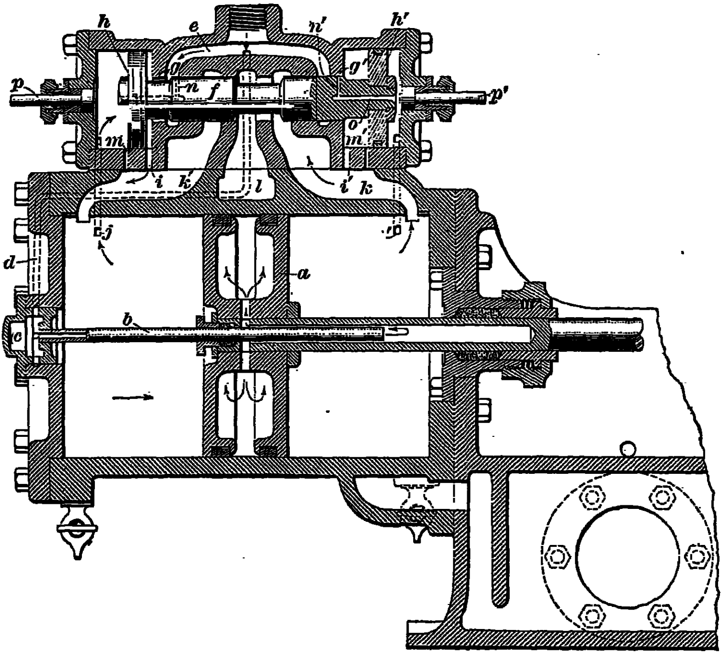


FIG. 5

between them, each section being provided with a packing ring. Steam, at boiler pressure, is admitted between the two parts of the piston by means of the tube *b*, which is rigidly secured to, and is in communication with, the chamber *c*. A steam-tight joint between the tube *b* and the piston *a* is made by a stuffing-box and gland, to prevent leakage into the main steam cylinder. A small port *d*, shown by dotted lines, supplies steam to the chamber *c*. When the piston is moving to the right, steam

enters from the space *e* through the annular opening between the reduced neck of the valve *f* and bore of the left chest wall *g*. The steam is thus projected against the inside surface of the valve head *h* before escaping through the port *i* and passing into the cylinder. Both the pressure and the impulse due to the velocity of the entering steam act on the valve head *h* and force it to the left, thus tending to close the annular opening in the chest wall *g*. The steam flowing through the annular opening and the port *i* into the cylinder also flows through the small port *j* to the left of the valve head *h*. The steam entering through these ports is wiredrawn, so that its pressure is reduced, but it acts on a greater area of the valve head *h* than the steam on the right of *h*. Hence, the valve *f* moves to a position where the total forces acting on the two sides of *h* are equal, and the valve then remains stationary. The steam entering through the annular opening in the chest wall *g* is also wiredrawn, so that the pressure on the left of the piston *a* is below the full boiler pressure existing in *e*.

While the piston *a* is moving to the right, the steam on the right is exhausting through the port *k* into the exhaust port *l*. As the piston runs over the port *k*, the port *j'* leading to the right of the valve head *h'* communicates with the space within the piston containing steam at boiler pressure, and this live steam rushes into the space to the right of the valve head *h'*. Since the steam pressure on the left of *h* is less than the pressure on the right of *h'*, the valve moves to the left, and by doing so closes the left steam inlet, opens the left exhaust, and also opens the right steam inlet in the chest wall *g'* and closes the right exhaust. The live steam admitted to the right of the piston *a* first brings it to rest and then reverses its motion. Obviously the piston *a* will cover the ports *j*, *j'* for a short time after beginning each stroke and thus shut off the admission of live steam to the left of *h* and the right of *h'*. In order that this closing of the ports *j*, *j'* may not affect the position of the valve *f*, auxiliary steam ports *m*, *m'* connect the chambers containing the valve heads with the main steam ports *k*, *k'*.

The valve *f* has two small ports *n*, *n'*, called *preadmission ports*, that connect with axial passages, as *o*, leading to the left

of h and to the right of h'. These preadmission ports are so placed that when the valve f is in its mid-position their centers will be in line with the inner edges of the chest walls g and g'. The purpose of the preadmission ports is to admit live steam to the left of h or the right of h' in case the valve f comes to rest near its mid-position on stopping the pump. Should the valve come to a stop exactly in its mid-position, both preadmission ports n, n' will be uncovered to live steam, and it is then necessary to shift the valve by hand. This is done by pushing in either one of the tappets p, p', which are also used when the valve is stuck from any cause.

15. The valve motion of the Marsh pump automatically adjusts the live-steam pressure on the steam piston to suit the resistance at the water end by increasing or decreasing the port opening for the admission of the live steam, at the chest walls g, g', Fig. 5. As previously stated, the live steam entering the cylinder is wiredrawn, so that it is somewhat below the steam-chest pressure. If the resistance at the water end is increased, the steam-end piston will slow down, with the result that the steam pressure within the cylinder is increased. This is due to the fact that there is less wiredrawing of the steam at the lower speed, and the increase of pressure is communicated through the port j or j' to the left of h or the right of h', and thus shifts the valve f to give a greater port opening at the chest walls g or g' to meet the increased resistance. In a similar manner, if the resistance on the water piston is reduced, the port opening at the chest walls will be decreased and the steam pressure in the cylinder will be further reduced by wiredrawing.

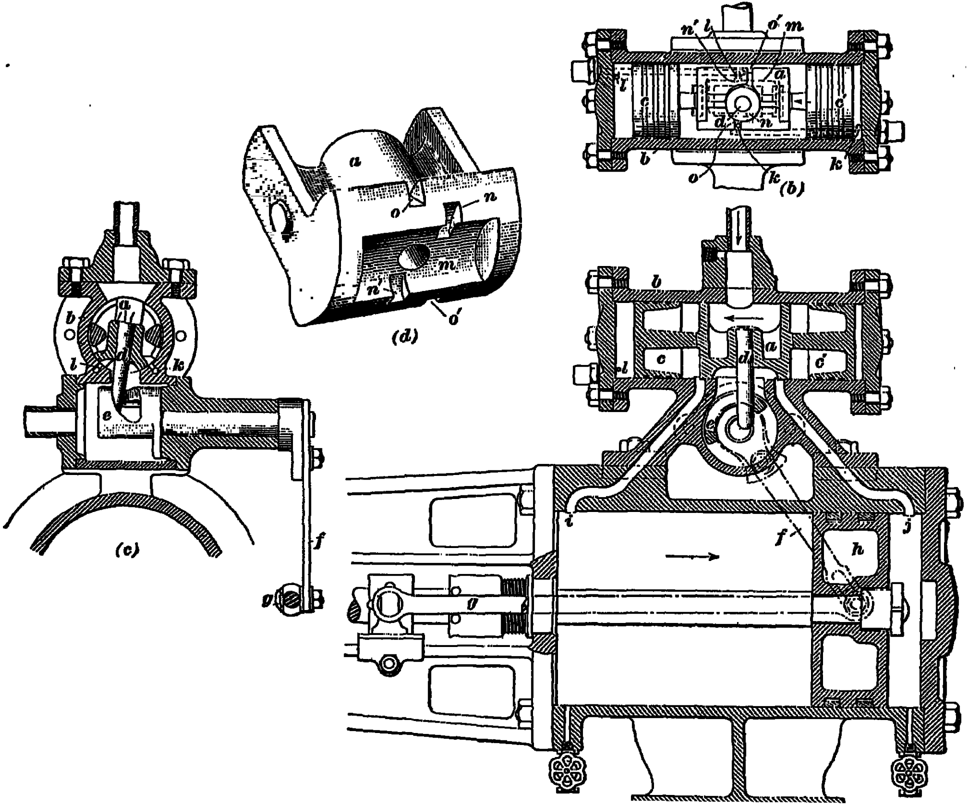
In a modification of the design shown in Fig. 5, the tube b is omitted and the piston a is made in the form of a spool, with the heads separated so far that when the piston is at the end of the stroke the one head is still a slight distance from the middle of the cylinder. The passage d then connects with the center of the cylinder, thus supplying steam at steam-chest pressure at all times to the space between the piston heads. This design involves the use of a much longer cylinder for a

given stroke than the one in Fig. 5, but it gives a simpler construction.

16. In Fig. 6 is shown the steam end of the **Davidson** single direct-acting pump, built by M. T. Davidson, Brooklyn, New York. In this pump the valve that distributes the steam is actuated both by a positive connection to the piston rod and by steam pressure. The view (*a*) is a vertical section of the cylinder and steam valve; (*b*), a horizontal section of the valve chest; (*c*), a vertical section through the valve and chest at right angles to the center line of the cylinder; and (*d*), a perspective view of the valve.

The valve *a* is in the valve chest *b*, between the two pistons *c*, *c'*. A pin *d* fixed in the body of the valve extends downwards and enters a slot in the cam *e*. This slot has a curved edge that bears against the pin *d*, as shown in (*c*). Consequently, as the piston moves to and fro, the links *f*, *g* transmit motion to the cam *e*, causing it to oscillate back and forth, and the curved edge of the slot in turn acts on the pin *d* and causes the valve *a* to oscillate. In view (*a*), the link *f* and part of the link *g* are shown in dash-and-dot lines. In reality, these parts lie in front of the plane on which the section is taken, and could not, therefore, be seen. This renders it necessary to adopt a conventional method of illustration, which, in this case, consists of showing the parts by dash-and-dot lines. Besides the oscillating motion, the valve *a* also has a reciprocating motion longitudinally in the valve chest *b*. This latter motion is imparted to it by the movement of the pistons *c*, *c'*, behind which live steam is alternately admitted through the ports *k*, *l*.

Assuming the piston *h* to start at the middle of its stroke and to move to the right, the curved edge of the cam *e* gradually forces the pin *d* to the side, turning the valve *a* into the position shown in (*c*) when the piston reaches the position shown in (*a*). With the piston *h* in this position and moving in the direction of the arrow, the end of the slot in the cam *e* bears against the pin *d*, and further movement of *h* to the right causes the slide valve *a* to be pushed to the left, thus closing the main ports *i* and *j*. This closing of the port *j* causes some steam to

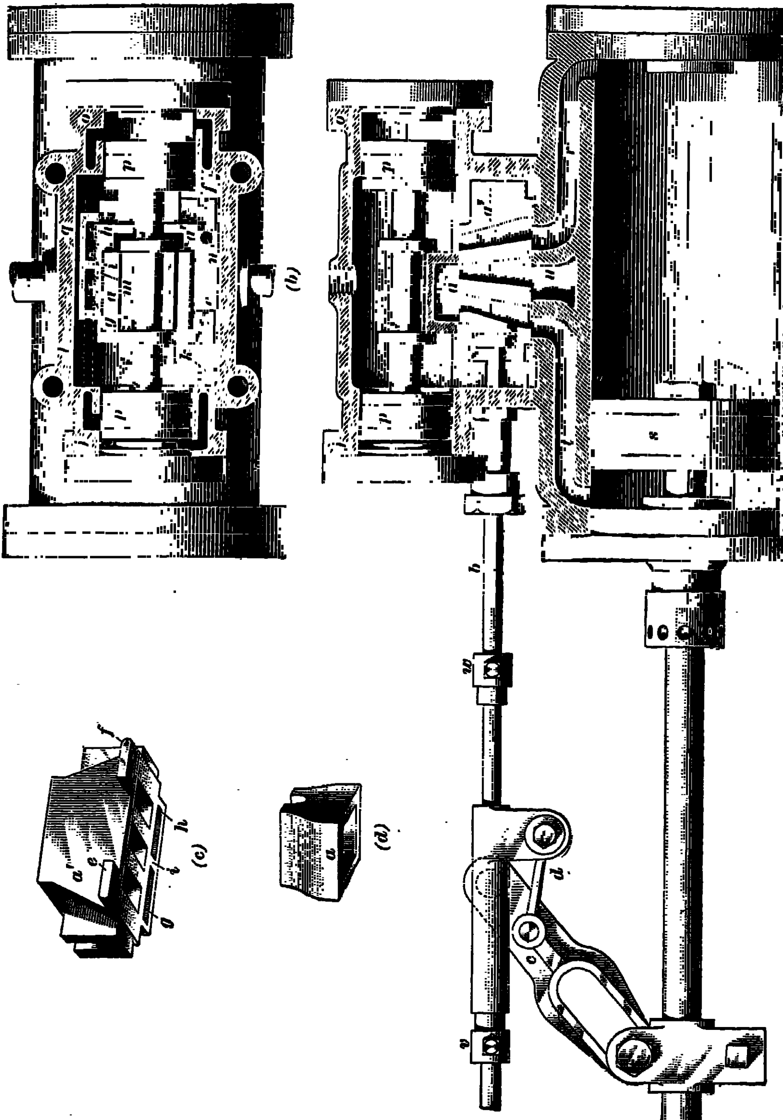

(b)
(c)
(d)

be trapped between the main piston *h* and the right cylinder head, and thus cushions the piston.

17. In the valve chest *b*, Fig. 6, are two ports *k* and *l* connected by passages to the two ends of the chest, as shown in (*b*) and (*c*). The valve *a* has beneath it an exhaust cavity *m*, with two slots *n*, *n′* extending from the cavity toward the outer edges of the valve, as shown in (*d*). On the outer edges of the valve are two slots *o*, *o′* extending inwards from the outer edges. The partial rotation of the valve, due to the motion of the piston from the middle of its stroke, causes the valve to assume the position shown in (*c*). As soon as the valve *a* is shifted by the piston movement so as to cover the ports *i* and *j*, that is, to stand in its central position, the port *k*, in *b*, is uncovered by the slot *o* of the valve, thus admitting steam to the right of *c′*, and the port *l* is connected to the exhaust cavity by the slot *n′*. Since live steam is thus admitted to the right-hand side of *c′*, and the space at the left-hand end of *c* is connected to the exhaust, the greater pressure on the right of *c′* forces it to the left, carrying with it the valve *a* and the piston *c*. This movement opens the main port *j* to live steam and the port *i* to exhaust, whereupon the piston *h* at once travels to the left.

On passing the mid-position in the direction opposite to that shown by the arrow, the motion of the piston *h* causes the cam *e* to throw the pin *d* to the right instead of to the left, as shown in (*c*). The ports *k*, *l* are then covered by the valve and the slight longitudinal motion given to the valve during the last part of the stroke of the piston opens the port *l* to live steam through the slot *o′* in the valve, and the port *k* is connected with the exhaust through the slot *n*. Since the live steam is then acting on the left of *c*, and since the right of *c′* is open to exhaust, the valve *a* and the two small pistons *c*, *c′* are forced to the right, opening the main port *i* to live steam and *j* to the exhaust, whereupon the piston *h* again moves to the right.

From the description, it is apparent that the valve *a* is given a rocking motion by the cam *e* for the purpose of making the auxiliary ports *k*, *l* serve alternately for live steam and exhaust.



(b)
(c)
(d)

A positive mechanical closure of the main steam ports is provided shortly before the end of the stroke to prevent the main piston *h* from striking the cylinder heads, and the ports *i*, *j* are opened by steam pressure applied alternately behind *c* and *c'*.

18. In Fig. 7 are shown views of the steam end of a **Blake** single direct-acting pump, built by what is now the Blake-Knowles Steam Pump Works, East Cambridge, Massachusetts. View (*a*) is a sectional elevation, (*b*) a plan view, (*c*) a perspective view of the movable seat, and (*d*) a perspective view of the main valve. In this pump the main valve *a* slides on a movable valve seat *a'*, which derives its motion from the piston rod through the rod *b*, the forked lever *c*, and the link *d*. The forked lever *c* engages a roller carried on the crosshead fastened to the piston rod and is pivoted at its upper end on a standard fastened to the frame of the pump and not shown. The valve seat *a'* is provided with two lugs *e*, *f*, and has two exhaust cavities *g*, *h* that are separated by the partition *i*. At *j* and *o* are two auxiliary cylinders to which live steam is admitted alternately through the ports *k*, *n*, and from which the steam is exhausted through the ports *l*, *q*. In these cylinders, the pistons *p* move back and forth, carrying with them the valve *a*. In the position in which the parts are shown, live steam is cut off from the auxiliary cylinder *j* by the lug *e*, which covers the inlet to the auxiliary live-steam port *k*, shown in dotted lines; the cylinder *j* is open to the exhaust through the ports *l*, *m*, which are connected by the exhaust cavity *g*; and the port *m* connects directly with the main exhaust. The auxiliary live-steam port *n* for the cylinder *o* is uncovered by the lug *f*, and the live steam enters this cylinder, and pressing against the part of the double piston *p* that is in *o*, holds the piston *p* and the main valve *a* in its extreme left position, as shown. The steam in the cylinder *o* cannot exhaust at present through the auxiliary exhaust port *q*, because the partition *i* separates *q* from the port *m*.

Steam entering the main cylinder through the port *r* is forcing the main piston *s* to the left, the left end of the cylinder being open at present to the exhaust through the ports *t*, *u*.

When the piston *s* reaches the position shown, near the end of the stroke, the sliding tappet to which the link *d* is hinged strikes the collar *v* fastened to *b*, and carries the valve seat *a'* to the left. By this motion, the lug *e* uncovers the auxiliary live-steam port *k*, admitting live steam behind the piston *p* in *j*, and the lug *f* covers the auxiliary live-steam port *n*. At the same time, the partition *i* passes to the left of the exhaust port *m*, whereby the steam in *o* can now exhaust through *q* and *m*, while communication of the auxiliary exhaust port *l* with *m* is interrupted. The live steam in *j* now shifts the pistons *p*, and hence the main valve *a*, to their extreme right-hand positions, thereby admitting live steam through *t* and permitting exhaust through *r* and *u*. When the piston *s* is near the end of the right-hand stroke, the sliding tappet on *b* engages the collar *w* and forces the seat *a'* to the right. In this way, steam is admitted to *o* and exhausted from *j*, thus forcing *a* to the left and admitting live steam through *r* and permitting exhaust through *t* and *u*, which causes the piston *s* to move to the left again.

The auxiliary pistons *p* are cushioned by some exhaust steam that is retained in the cylinders *j*, *o*. The main piston *s* is cushioned by live steam entering before the end of the stroke. The length of the stroke of the main piston *s* is regulated by adjusting the distance between, and the position of, the collars *v* and *w* on the rod *b*, thereby determining the extreme right and left positions of the valve seat *a'*.

19. The steam end of a **Dean Brothers** single direct-acting steam pump, built by the Dean Brothers Steam Pump Works, Indianapolis, Indiana, is shown in Figs. 8 and 9. Fig. 8 (*a*) is a perspective view of the steam end with some of the external parts cut away; view (*b*) shows the auxiliary valve and valve seat in perspective, with the auxiliary valve removed some distance from its seat, and view (*c*) shows in elevation the auxiliary valve seat and the passages leading from it. Fig. 9 is a perspective cross-sectional view of the steam end.

20. The operation of the steam end is briefly as follows: The passage of the steam to and from the main steam cylinder *a*,

Fig. 9, is controlled by the slide valve *b*, which is moved by the chest piston *c*; this is moved by steam which, through suitable ports, is admitted to and exhausted from the chest-piston cylin-

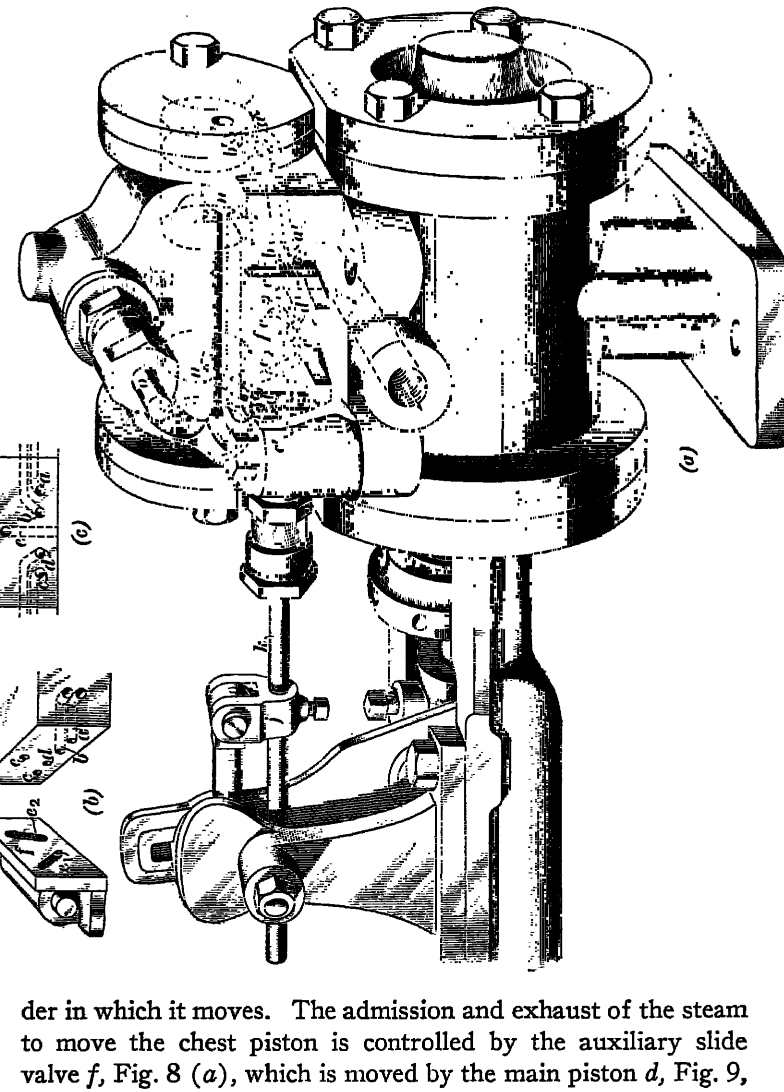


Fig. 8

der in which it moves. The admission and exhaust of the steam to move the chest piston is controlled by the auxiliary slide valve *f*, Fig. 8 (*a*), which is moved by the main piston *d*, Fig. 9,

through a rocker-arm pivoted to a bracket and connected by a hinged rod to a collar *j*, Fig. 8 (*a*), fastened to the valve stem *k* of the auxiliary valve *f*.

21. The main slide valve *b*, Fig. 9, is an ordinary D slide valve, which, in the position in which it is shown, is admitting

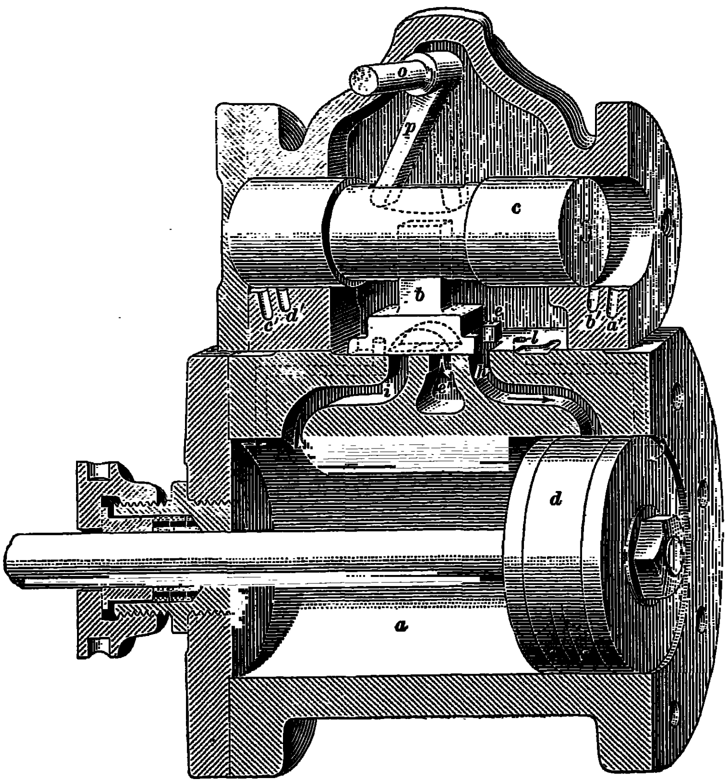


FIG. 9

steam to the port *h* and permits exhausting from the left-hand end of the main cylinder *a* through the ports *i* and *e'*. Since the piston *d* has covered the lower end of the port *h*, no steam can enter the main cylinder from that port. There is, however, an auxiliary port *l* which has been uncovered by the projection *e* on one corner of the main valve *b*; a small quantity of

steam is thus admitted on the right-hand side of the piston *d* to start it slowly.

22. The chest in which the auxiliary valve *f*, Fig. 8 (*a*), is placed is at one side of the main valve, and communicates with the main steam chest through holes *n*, *n* drilled in such a position that at least one of them is always uncovered; high-pressure steam therefore always fills the auxiliary valve chest. The seat for the auxiliary valve, as shown in Fig. 8 (*a*), (*b*), and (*c*), has five ports *a*, *b*, *c*, *d*, and *e*; the ports *a*, *b*, *c*, and *d*

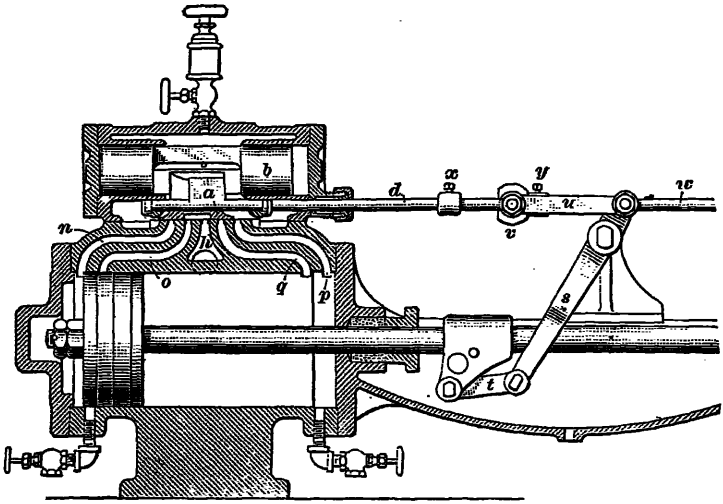


Fig. 10

connect with the ends of the chest-piston cylinder through the passages *a'*, *b'*, *c'*, and *d'*, shown both in Figs. 8 (*a*) and 9. The port *e*, Fig. 8, connects with the exhaust passage of the pump. When the auxiliary valve *f* is in the position shown in Fig. 8 (*a*), the port e_1 in its face connects the ports *d* and *e* in the seat and the steam at the left of the chest piston is allowed to pass into the exhaust; the port e_1 can be clearly seen in view (*b*). The port *a*, Fig. 8 (*b*) and (*c*), is uncovered by the right-hand edge of the auxiliary valve *f* at about the same time the exhaust from the left of the chest piston is opened, and hence the chest piston and main valve are moved to the

left by the high-pressure steam passing through the passage *a′*. When the pump piston has reached the left-hand end of its stroke, the auxiliary valve *f*, Fig. 8 (*a*) and (*b*), has moved to the right, thereby connecting the ports *b* and *e* by means of the port e_2 in its face; this port can be clearly seen in (*b*). The auxiliary valve *f* has also uncovered the steam port *c*, and consequently the chest piston is driven to the right.

23. Exhaust from the chest-piston cylinder takes place through the passages *b′*, *d′*, Figs. 8 (*a*) and 9, which are covered by the chest piston before it reaches the end of its

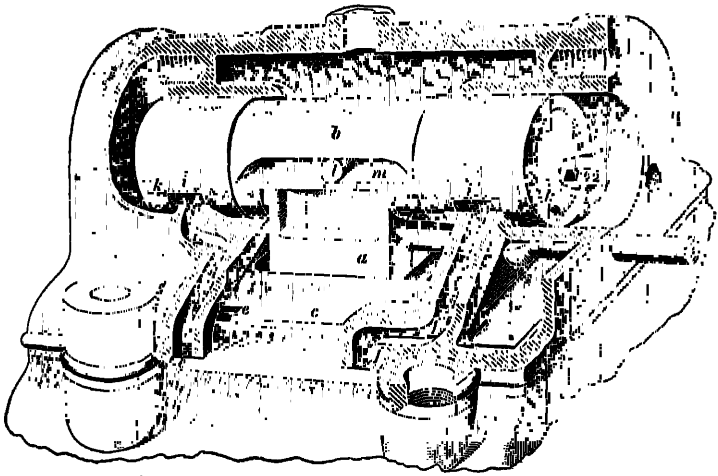


FIG. 11

stroke in either direction. The steam thus trapped in the ends of the chest-piston cylinder is compressed by the chest piston, which is thus quietly brought to rest and also prevented from striking the heads of the chest-piston cylinder.

A shaft *o*, Figs. 8 (*a*) and 9, provided with a handle which is not shown, carries a lever *p*, Fig. 9, by means of which the chest piston and hence the main valve can be moved by hand to warm up the pump cylinder and to start the pump.

24. The steam end of a **Deane of Holyoke** single direct-acting steam pump, built by the Deane Steam Pump Company, Holyoke, Massachusetts, is shown in Figs. 10 to 14, in all of

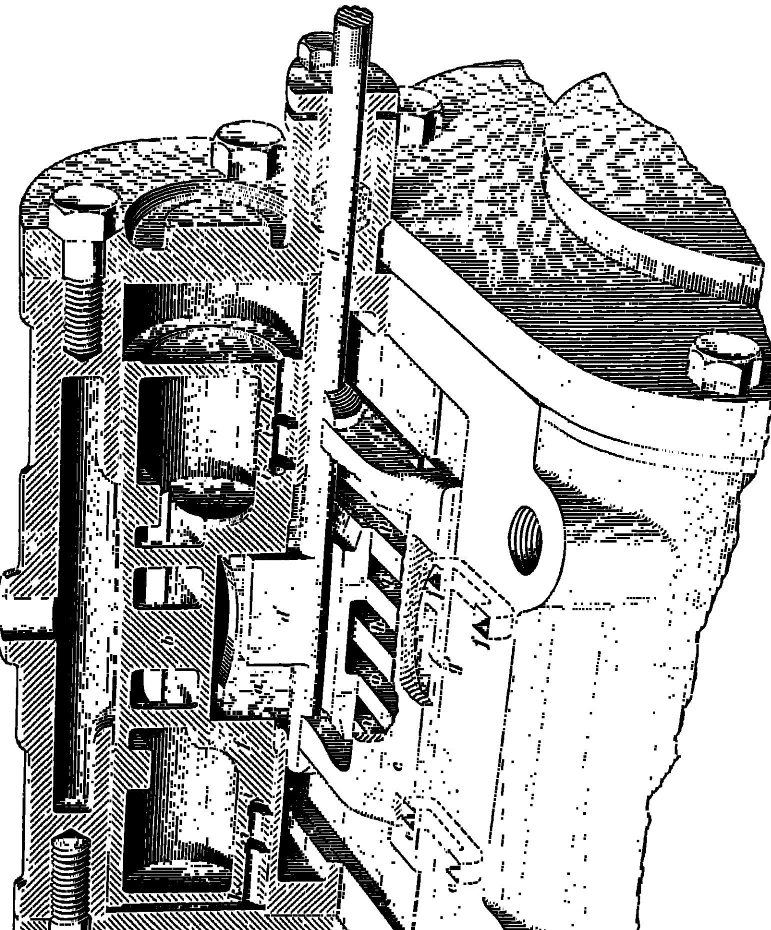


FIG. 12

which the same parts have been given the same reference letter. Fig. 10 is a general view of the steam end in section; Fig. 11 is a perspective view of the valve chest and valves with some of the external parts cut away to show some of the ports and internal parts to better advantage. Fig. 12 is another perspective view of the steam end, with the front half of the valve chest, chest piston, and slide valve, but only a small part of the auxiliary valve, cut away. Fig. 13 is a perspective view of the chest piston removed from the valve chest, and Fig. 14 a diagrammatic plan of the valve chest and ports.

25. The operation of the pump is as follows: The auxiliary slide valve *c*, Fig. 11, is moved by the main piston through the lever *s*, Fig. 10, the links *t* and *u*, and the sliding collar *v* which strikes alternately the collars *x* and *y* fastened to the valve stem *d*. The movement of the auxiliary valve *c*, Figs. 11, 12, and 14, admits steam to the chest cylinder, thereby moving the chest piston *b*, Figs. 10 to 14, and with it the main slide valve *a*, Figs. 10 to 12. The main slide valve *a* admits live steam to the pump cylinder either through the port *n* or the port *p*, Figs. 10 and 12, which two steam ports open into the ends of the pump cylinder so that the main piston cannot cover them. Steam is exhausted from the pump cylinder through the ports *o* and *q*, Figs. 10 and 12, which are covered by the pump piston before the end of the stroke, thus retaining steam to cushion the piston. The seat for the main valve also contains the exhaust port *h*, Figs. 10 and 12, which connects to the exhaust pipe.

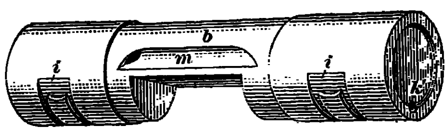
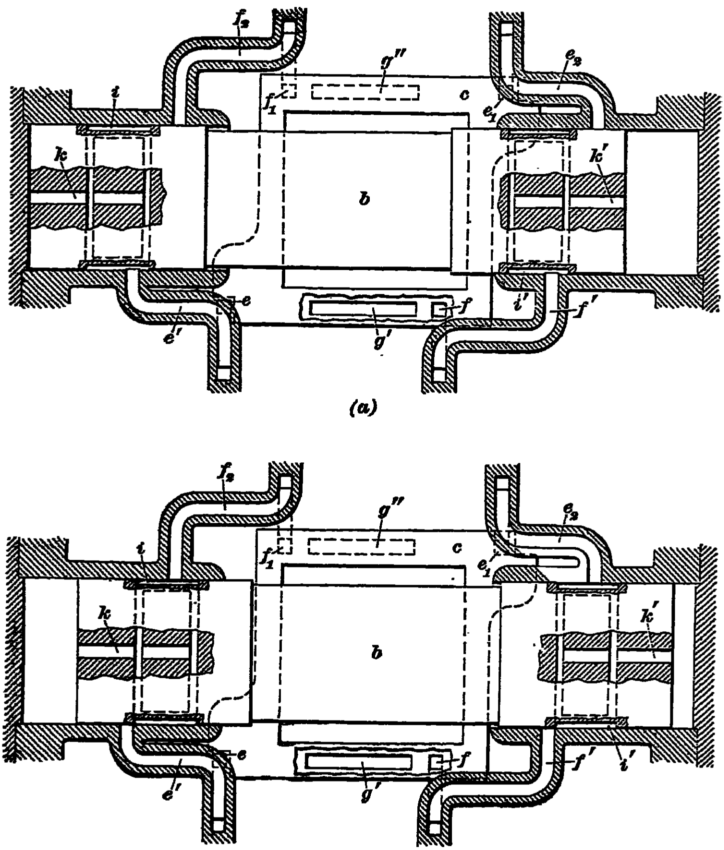


FIG. 13

26. The main steam valve *a*, Figs. 10 to 12, is an ordinary **D** slide valve with two projections on its top side that fit into a slot on the lower side of the chest piston *b*; this slot can be clearly seen in Fig. 13. The auxiliary valve seat is divided into two parts, one on each side of the main valve. The auxiliary valve *c*, Figs. 11, 12, and 14, is approximately a hollow rec-

tangle, and surrounds the main valve. The port *e* in the auxiliary valve seat, Figs. 11, 12, and 14, connects with a passage *e′*, Figs. 11 and 14, in the wall of the steam chest, and, when the chest piston is in the position shown in Figs. 11, 12,



(a)

(b)

Fig. 14

and 14 (*a*), it connects also with the port *i* (also see Fig. 13) in the chest piston and thereby with the hole *k*, Figs. 12 and 14, drilled into the chest piston. A similar hole *k′*, Figs. 11 to 14, connects the space to the right of the chest piston to the port *i′*

in the chest piston, and thence by way of the passage f', Figs. 11 and 14, to the port f in the auxiliary valve seat. A port e_1 and passage e_2, Fig. 14, and a port f_1 and passage f_2 serve the same purpose as the ports e, f and passages f, f'. The ports g', Figs. 12 and 14, and g'', Fig. 14, in the auxiliary valve seat connect with the exhaust cavity of the pump.

27. When the auxiliary valve c, Fig. 14, has been forced into the position shown, the port e is uncovered, thus admitting high-pressure steam to the left of the chest piston b. At the same time the space at the right of the chest piston opens into the exhaust cavity, as the ports f and g' are connected by a recess or port in the lower face of the auxiliary valve; this recess can be clearly seen in Fig. 12. The high-pressure steam now drives the chest piston and main valve to the right, thus admitting steam to the left of the main piston and opening the exhaust port h, Fig. 12. Referring again to Fig. 14, as soon as the chest piston has moved to the position shown in (b) the exhaust port f' is closed, and the steam trapped in the space at the right of the chest piston b cushions this piston. Just before the main piston reaches the end of its stroke to the right, the auxiliary valve c is shifted to the left, thereby uncovering the steam port e_1 and connecting the ports f_1 and g''. Live steam then passes to the space at the right of the chest piston b, driving this as well as the main valve to the left, thereby admitting live steam to the right of the main piston.

28. Rotation of the chest piston is prevented by a pin l, Fig. 11, in the wall of the chest-piston cylinder engaging a slot m, Figs. 11 and 13, in the chest piston b. Should the chest piston fail to shift when the auxiliary valve admits live steam to the chest-piston cylinder, a block d', Fig. 12, on the valve stem d of the auxiliary valve will strike the chest piston and move it as well as the main valve before the main piston reaches the end of its stroke.

29. Two sectional views of the steam end of a Style A **Blake-Knowles Simplex** steam pump, built by the Blake-Knowles Steam Pump Works, East Cambridge, Massachusetts, are shown in Fig. 15. The valve motion consists of an auxiliary

slide valve *a* riding on top of the main valve *b*, and a chest piston *c* that is thrown to the right or left by steam admitted to its cylinder by the auxiliary valve *a*; the chest piston *c* carries the main valve *b* with it whenever it moves. The chest piston *c* and main valve *b* are placed, and move, at right angles to the pump cylinder *d*; the auxiliary valve *a* is in line with the pump cylinder and is operated by the pump piston *e* through the rocker-arm *f* carrying the tappet *g*, which tappet engages one or the other of the collars *h*, *h'* on the valve stem of the auxiliary valve. There is one steam port *i* and one port *j* at each end of the pump cylinder, all four leading to the seat of the main valve; when viewed from the top the two ports leading to one end of the pump cylinder lie on its right side and those leading to the other end lie on the left side, and therefore only the one set of ports can be seen in Fig. 15 (*a*), although the one port *j'* can be seen in section in (*b*). Auxiliary ports *k*, *k'* lead

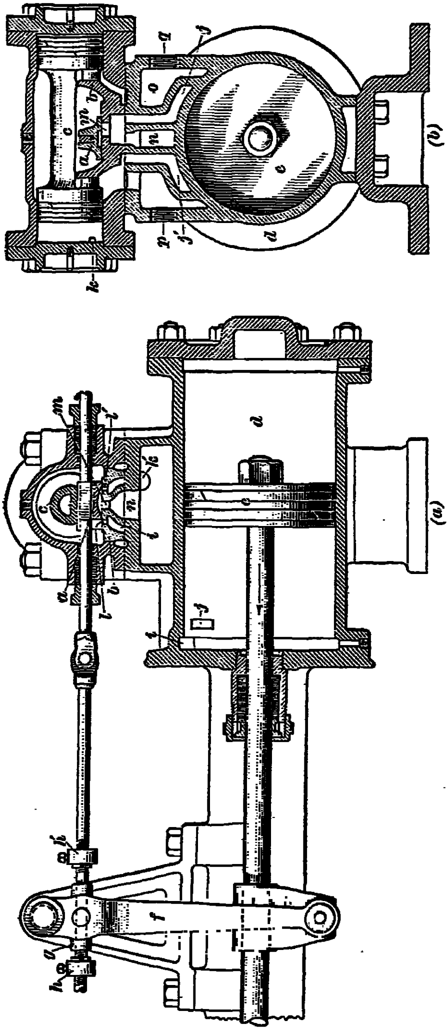


Fig. 15

to the two ends of the chest-piston cylinder from the seat of the main valve, and serve alternately as steam and exhaust ports; steam is admitted to, and exhausted from, the right or left of the chest piston by the auxiliary valve *a* alternately opening and closing ports *l*, *l'* in the main valve *b* or connecting them to the exhaust port *m* of the main valve *b*. The exhaust port *n* through the main-valve seat connects to the exhaust cavity *o* of the pump cylinder. Live steam is admitted to the steam end at *p*, and the exhaust pipe is connected at *q*.

The operation is as follows: With the piston *e* moving to the left, as shown by the arrow, the auxiliary valve *a* has uncovered the port *l* of the main valve and placed the ports *l'* and *m* in communication; therefore, live steam has passed through the auxiliary port *k* to the left of the chest piston *c* while the right is open to the exhaust by way of the ports *l'* and *m*. The chest piston *c* and the main valve *b* are therefore held at the right in the position shown in (*b*), and as the main valve has uncovered the port *j'* as well as the port corresponding to *i*, live steam flows into the right-hand side of the pump cylinder *d*, driving the piston *e* to the left. At the same time the main valve has placed the port *j* in communication with the exhaust cavity *o* by way of the port *n*, so that steam can exhaust from the left of the piston *e*; the port *i* is closed by the main valve, but communicates with the port *j* by means of a small drilled hole known as a *cushion relief hole*.

As soon as the piston *e* covers the port *j*, steam can exhaust from the left only by way of part of the small port *i*, through the cushion relief hole, into the port *j*, and thence to the exhaust pipe; owing to the small size of the cushion relief hole the passage of the exhaust steam from the left of the pump piston is so restricted that the piston is cushioned thereby, that is, brought quietly to rest. When the piston *e* has nearly completed its stroke to the left, the tappet *g* engages the collar *h* and pulls the auxiliary valve *a* to the left, thereby putting the main-valve ports *l* and *m* into communication and opening the main-valve port *l'*. This admits live steam to the right of the chest piston *c* and at the same time permits exhaust from the left of the chest piston *c* through the passage *k* by way

of the ports *l*, *m*, and *n*, and the chest piston *c* is thus driven to the left, carrying the main valve *b* with it. The movement of the main valve closes the port *k*, and the steam thus confined at the left of the chest piston cushions it. The main valve has at the same time opened the ports *i* and *j* to live steam, and the port *j'* to the exhaust; consequently, the piston *e* now moves to the right.

DUPLEX DIRECT-ACTING STEAM PUMPS

30. A **duplex pump** consists of two similar pumps placed side by side and so connected that the operation of one has a definite relation to the operation of the other. The two are designed to work together, and in most duplex types it is difficult to run one pump without the other. Duplex pumps, like single pumps, are made either with pistons or with plungers. Pistons are preferred for moderate pressures and plungers for high pressures, and, usually, each pump is double-acting.

The two single pumps that make up a duplex pump are of similar construction. A lever attached to the piston rod of one pump operates the slide valve of the steam cylinder of the other. The effect of this arrangement is such that, when the piston or plunger of one pump arrives at a point between the middle and the end of its stroke, the location depending on the construction, the plunger or piston of the other begins its stroke, thus alternately taking up the load, producing a steady flow of water, and avoiding the stresses induced by a column of water when suddenly stopped or set in motion.

31. In Fig. 16 is shown the general arrangement of one design of a horizontal duplex pump, and in Fig. 17, a section through one side. The same parts have been given the same reference letters in both illustrations.

This type of duplex pump is built by many builders, and is generally known as the *Worthington duplex pump*, after the name of its inventor. The steam valves *a*, Fig. 17, of this pump are of the plain **D** type, with double ports *b*, *b'* and *c*, *c'* in the valve seat. The valve *a* of the front pump is operated by the rocker-arm *d*, which engages with the piston rod of the

with live steam. Hence, it is important to give the pump a long stroke. The water inlet *i*, Figs. 16 and 17, is usually located so as to bring the water up between the two pump cylinders.

The water plunger *j*, Fig. 17, is double-acting and works through a deep metallic ring *k* that is bored to fit the plunger accurately and bolted to the cylinder partition *k'*. The plunger is located above the suction valves *m* a suitable distance so as to form chambers *n* into which any foreign substances may fall clear of the wearing surfaces. The air chamber *o* is in direct line with the flow of water from the discharge valves *p*. When the water pressure is not higher than 150 to 200 pounds per square inch, either a plunger *j*, which works in a ring *k*, as shown, or a piston packed with fibrous packing, may be used in the water end. But if the pressure is higher, say more than 300 or 400 pounds per square inch, packed plungers are required in order to avoid excessive leakage.

33. Fig. 18 shows the steam end of a **Worthington piston-valve** duplex steam pump. The valves are operated in the same manner as those of the pump shown in Figs. 16 and 17, but the lost motion is obtained by a special construction of the valve rod *a*. This rod is divided into two parts. The part attached to the valve stem carries a slotted yoke *b*; the part attached to the crank is free to slide within the yoke and carries a collar *c* pinned to it. The collar *c* alternately strikes against the check-nuts *d* and *e* on the yoke *b* and then carries the valve with it. The lost motion is quite large, as the valve needs to be moved but a slight amount.

The length of stroke is adjusted by the use of the so-called *dash relief valves f*. These valves, which are simply pointed setscrews, control passages *i* connecting the steam ports *g* and exhaust ports *h*, and are set by trial to the correct position and then locked with the cap nuts *f'*. The action is as follows: When the piston on its exhaust stroke covers the port *h*, no further exhaust can take place, and the steam will be compressed between the piston and the cylinder head. The location of the ports *h* is so chosen that the compression will stop the piston just short of the cylinder head at the highest speed at

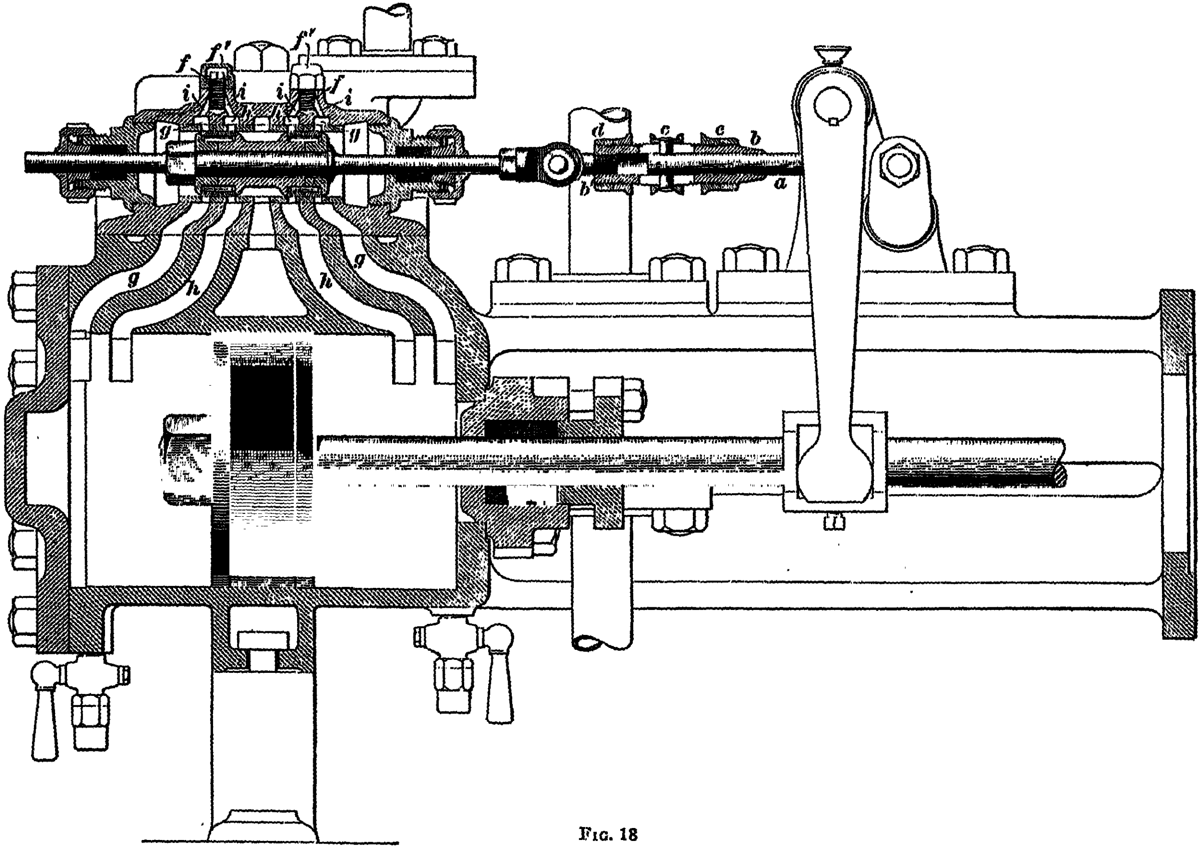


Fig. 18

which the pump can operate. It is evident that when the pump is working at slow speed, the compression being the same as at high speed but the momentum of the moving parts being less, the piston will stop earlier than at high speed; that is, the stroke is shortened. The dash relief valves prevent this shortening by providing an escape for the exhaust steam after the exhaust ports *h* are closed. It is thus seen that by them the amount of compression is regulated to suit the speed of the pump, and the length of the stroke is thus kept constant.

Dash relief valves are applied to cylinders over 14 inches in diameter, as a general rule, and are used with slide-valve pumps as well as with piston-valve pumps. In either case they simply control a passage by which the exhaust port and steam port communicate.

MULTIPLE-EXPANSION DIRECT-ACTING STEAM PUMPS

34. Neither in single nor in duplex simple direct-acting steam pumps is it possible to utilize the expansive force of the steam; while such pumps are very simple, reliable, and low in cost, they are extravagant in the use of steam. While the uneconomical use of steam is of relatively small importance with direct-acting pumps of small size, it becomes a serious matter with large pumps, which therefore, when economical operation is a controlling factor, are fitted with compound or triple-expansion steam ends.

35. Fig. 19 shows a common method of arranging the steam cylinders of a compound duplex direct-acting steam pump. The steam end for each half of the pump is made with two cylinders arranged tandem, the valves for both cylinders being driven from the same valve stem. The high-pressure cylinder is at the left and is connected to the low-pressure cylinder by a cast-iron yoke, or spacer *o*, which forms one head for each cylinder. The high-pressure piston rod passes through a sleeve in this spacer, as shown; this sleeve is held in its place by its flange being gripped between the spacer and a plate bolted onto the latter; otherwise, the sleeve is free to adjust itself slightly, being a free fit in the body. The exhaust passes

directly from the high-pressure cylinder through the pipe p to the steam chest of the low-pressure cylinder. Since there is no cut-off in either cylinder, the back pressure on the high-pressure piston is at all times equal to the pressure on the low-pressure piston, except for the resistance to the flow of steam through the ports and the pipe p.

Since the volume of steam admitted during each stroke is equal to the volume of the high-pressure cylinder, and this steam, when exhausted, just fills the low-pressure cylinder, it is evident that the number of expansions is equal to the ratio of the volume of the low-pressure cylinder to that of the high-pressure cylinder. Also, since the length of stroke is the same

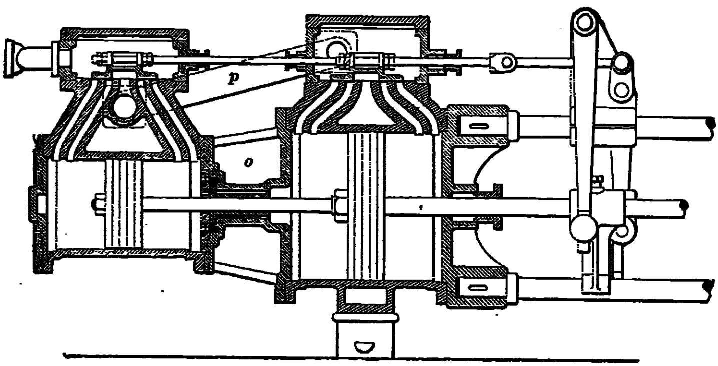


FIG. 19

for both cylinders, the number of expansions is equal to the ratio of the areas of the low- and the high-pressure piston. The usual number of expansions for small and medium sizes ranges from two to three. For large sizes four expansions are sometimes used.

Compound pumps are also made in which the cylinder arrangement is just the reverse from that shown in Fig. 19. In some of these compound pumps the high-pressure cylinder has no separate steam and exhaust ports; the compression and adjustment of length of stroke then takes place in the low-pressure cylinder.

36. In triple-expansion direct-acting steam pumps the arrangement shown in Fig. 20 is sometimes adopted for the

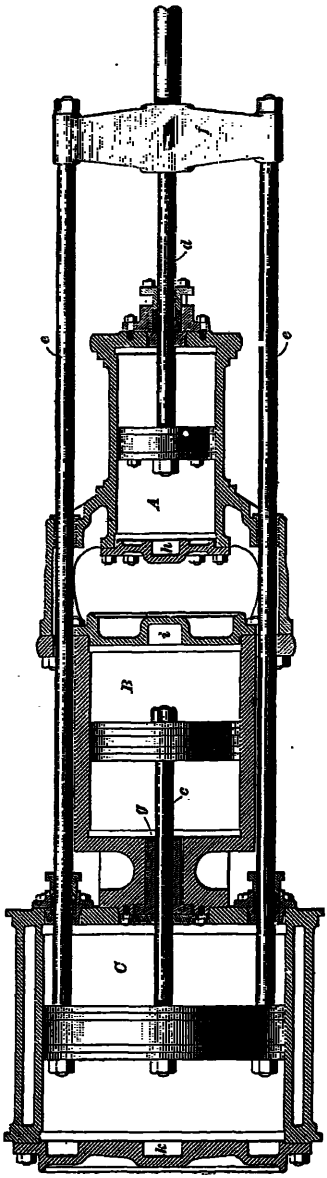


Fig. 20

steam end. This design makes all the pistons accessible and at the same time avoids the use of a stuffingbox between the high-pressure cylinder *A* and intermediate cylinder *B*. The low-pressure piston and intermediate piston are connected by the piston rod *c*, and the low-pressure piston is connected to the high-pressure piston rod by the side rods *e*, *e* and the yoke *f*. The piston rod *c* is nicely finished and ground and works through a cast-iron bushing *g*, which is a nice fit. This bushing can move sidewise slightly so as to accommodate any want of alinement between the two cylinders. At the same time it prevents leakage of steam from the intermediate cylinder *B* to the low-pressure cylinder *C*. The low-pressure and high-pressure stuffingboxes are quite accessible. Access to the different pistons is had by removing the covers *h*, *i*, and *k*.

37. Fig. 21 is a vertical longitudinal section of the pump whose piston rod and cylinder arrangement is shown in Fig. 20, and shows the steam distribution in this particular design of pump. In the illustration, *A* is the high-pressure cylinder; *B* is the intermediate

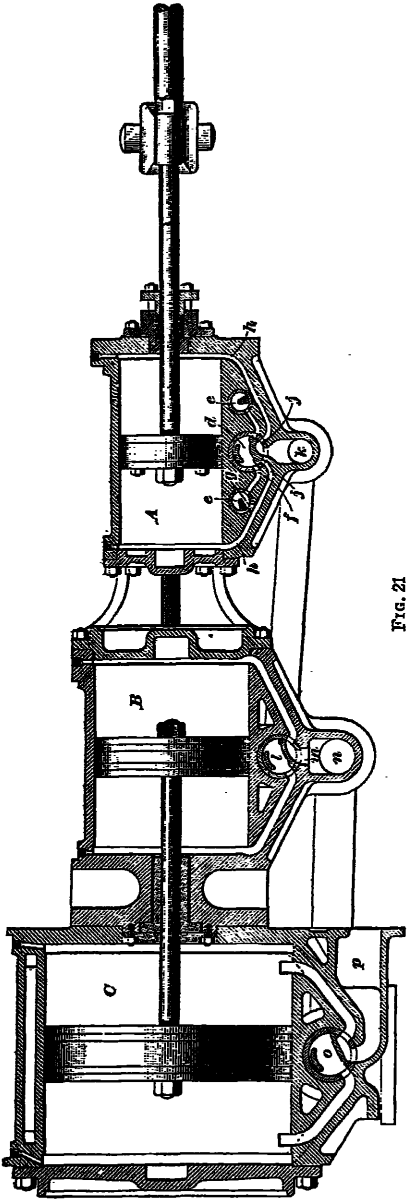


Fig. 21

pressure cylinder; *C* is the low-pressure cylinder; *d* is the high-pressure distributing valves; and *e, e* are the high-pressure cut-off valves. Steam enters through the center of the valve *d* and passes through the port *f* and through the cut-off port *g* into the high-pressure cylinder by way of the port *h*. The cut-off is effected by turning the rotary valves *e, e*. Exhaust from the high-pressure cylinder takes place through the ports *j, j* and thence into the high-pressure exhaust *k*, which leads to the inside of the intermediate steam valve *l*. The valve *l* is a rotary valve designed to distribute the steam exactly in the same manner as a common **D** slide valve. The intermediate- and low-pressure cylinders are not provided with cut-off valves. The exhaust steam from the intermediate cylinder passes out through the port *m* into the exhaust pipe *n*, and thence to the center of the low-pressure distributing valve *o*. From the low-pressure cylinder the steam is exhausted into the exhaust chest *p* and

thence into the condenser or atmosphere. Dash relief valves, not shown in the illustration, are provided on the low-pressure cylinders only. The distributing valves are worked as usual from the pump on the opposite side, while the cut-off valves are worked from the pump on which they are placed

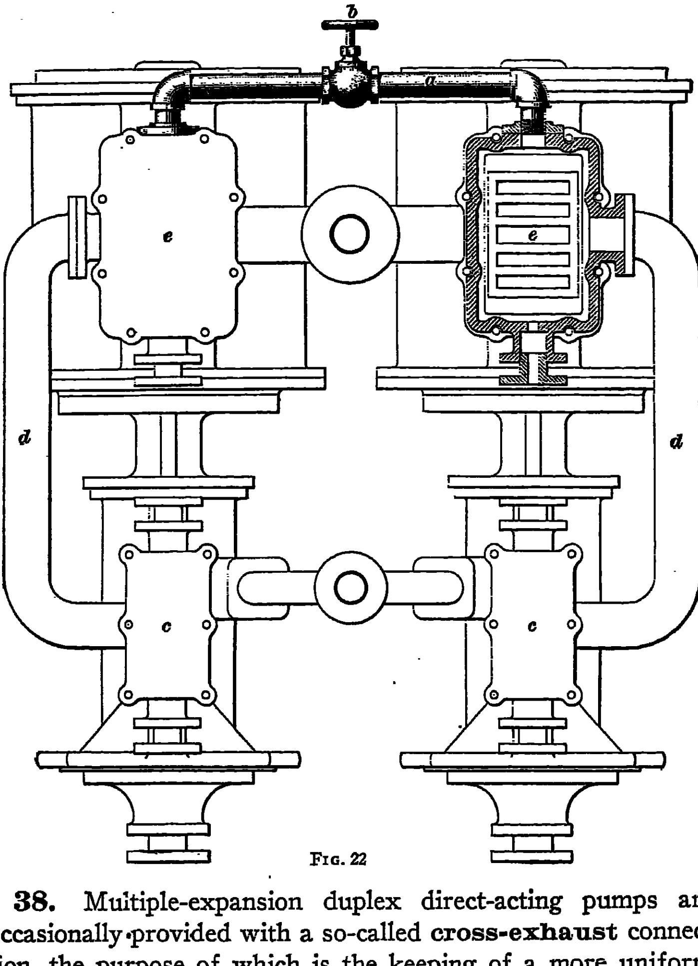


FIG. 22

38. Multiple-expansion duplex direct-acting pumps are occasionally provided with a so-called **cross-exhaust** connection, the purpose of which is the keeping of a more uniform pressure in the steam chests of the low-pressure cylinders than obtains otherwise. As shown in Fig. 22, it is simply a

pipe *a* of ample size, which is provided with a valve *b* and connects the steam chests of the low-pressure cylinders. The exhaust from the high-pressure cylinders *c*, *c* flows through the exhaust pipes *d*, *d* into the low-pressure steam chests *e*, *e*, but as the steam pressure there drops toward the end of the stroke, there is a diminishing of the impelling force on the steam pistons of the low-pressure cylinders that tends to shorten the stroke. With the valve *b* open, the exhaust from the high-pressure cylinder of one pump can pass to the low-pressure steam chest of the other pump just when the pressure in that steam chest commences to drop, and in consequence the pressure will be kept more uniform, which results in a steady and uniform motion.

HIGH-DUTY ATTACHMENT FOR DIRECT-ACTING STEAM PUMPS

39. The ordinary single or duplex direct-acting steam pump requires steam at full boiler pressure to be carried nearly to the end of the stroke, and consequently is very uneconomical in its use of steam. The economy of the direct-acting steam pump is greatly improved by making it compound or triple expansion, but even then it is not possible to secure the high ratios of expansion which are necessary for extreme economy in the use of steam, and hence of fuel, and which are demanded in large pumping plants for commercial reasons. In the ordinary steam engine, and also in the flywheel pattern of pump, power is stored up in the flywheel at the beginning of the stroke and given out when expansion begins, in order to have a uniform turning of the engine shaft, or a nearly uniform force acting upon the water piston in case of a pump. In a direct-acting steam pump, however, there are no heavy moving parts similar to a flywheel, and hence ordinarily no uniform impelling force can act on the water piston if steam is cut off early in the stroke. This defect led to the design of the **high-duty attachment,** which is simply a device that stores up power during the first half of the stroke and gives it out again during the second half, thus allowing steam to be used expansively in the steam cylinders.

The high-duty attachment in actual use was designed by Mr. J. D. Davies in 1879 and taken up and perfected by Henry R. Worthington. It is shown in Fig. 23 applied to a compound direct-acting pumping engine fitted with Corliss valves and cutting off early in the high-pressure and low-pressure cylinders. The piston rods are arranged so as to avoid internal stuffingboxes, and, in consequence, the pistons are accessible without having to dismantle the pump.

The two piston rods of the low-pressure piston and the high-pressure piston rod are attached to a common crosshead *a*, which runs in guides between the pump chambers and high-pressure cylinders. On this crosshead and opposite to each other are semicircular recesses. On the guide plates are cast two journal-boxes, one above and the other below the crosshead, equally distant from it and at the point equal to the half stroke of the crosshead. In these journal-boxes are hung two short cylinders *b* on trunnions that permit the cylinders to swing backwards and forwards in unison with the motion of the plunger crosshead. Within these swinging cylinders are plungers *c*, which pass through a stuffingbox on the end of the cylinders, and on the outer end have a rounded projection *c'*, which fits in the semicircular recesses in the crosshead. Consequently, as the crosshead moves back and forth, it carries with it the two plungers *c*, which, in turn, tilt the cylinders backwards and forwards. These swinging cylinders are called **compensating cylinders**; they are filled with water or with whatever fluid the pump may be handling. The pressure on the plunger within the compensating cylinders is produced by connecting the compensating cylinders through their hollow trunnions with an **accumulator** *d*, the ram of which moves up and down as the plungers of the compensating cylinders move in and out. The accumulator used is of the differential type; that is, it has a small cylinder *e* filled with oil or water in which its ram moves, and above it has a much larger cylinder *d* filled with compressed air. On the top of the ram of the accumulator is an enlarged piston rod carrying a piston which fits closely in the air cylinder. From this construction it follows that the pressure per square inch on the ram of the accumu-

lator will be the pressure of the air in the air cylinder per square inch multiplied by the ratio between the area of the air piston and that of the ram of the accumulator. The ratio of these areas is made to suit the particular service for which the pump is constructed. The pressure in the air cylinder is controlled by the pressure in the main delivery pipe of the pump, as it is connected to the air chamber *f* on the main delivery pipe.

40. The operation of the high-duty attachment is as follows: Suppose the pump is about to begin the forward stroke. At this time the compensating cylinders will be turned so as to point toward the steam cylinders, with their plungers at the extreme point of their outward stroke and at an acute angle with the line of motion of the crosshead, and with the full pressure of the accumulator load pushing them against the advance of the crosshead. As the pump plunger begins its forward stroke, each forward movement it makes changes the angle of the compensating plungers, until at mid-stroke the two plungers will stand exactly opposite each other and be at right angles with the pump plungers, which is the position in which they are shown in Fig. 23, in which position they can neither retard nor advance the movement of the pump plungers. Now, as the pump plunger passes the mid-stroke position, the compensating plungers begin to push the pump plunger along, whereas before and up to mid-stroke they resisted the movement of the pump plunger. This force increases constantly, until at the extreme end of the forward stroke, and when the compensating plungers are, as at the beginning, at their most acute angle, they exert their greatest force in helping to aid the pump plunger in its outward movement. The return stroke of the pump is made under precisely the same conditions as the forward stroke.

It is readily seen that at the beginning of the stroke and up to mid-stroke, work is being done in pushing the compensating plungers inward, and that after the crosshead passes the mid-position, work is being done by the compensating plungers. The effect of this is a nearly uniform force on the pump piston with a varying pressure in the steam cylinders.

41. An important feature connected with the use of the compensating cylinders is that the results obtained from their use are independent of the speed, in which respect their action is better than that of a flywheel.

FLYWHEEL STEAM PUMPING ENGINES

PURPOSE

42. Although direct-acting steam pumps cannot be excelled in simplicity, low first cost, and small expense for repairs, yet they can never be extremely economical in their use of steam, even when built compound and triple expansion. While there is little doubt that a high-duty attachment will greatly increase the economy, the fact remains that at present only a limited number thus fitted are in use.

In large pumping stations and in many other cases where the cost of fuel is of more importance than the advantages gained from direct-acting pumps, flywheel pumping engines are often used. These are steam engines with cranks and flywheels usually designed for the particular purpose of driving the pump to which they are attached. The steam valves are driven in the ordinary way by means of eccentrics; or some approved automatic valve gear may be used to operate them. By the use of the flywheel, steam may be cut off at the most economical point in the stroke, and the surplus energy imparted to the steam piston during the first part of the stroke will be stored in the flywheel, to be given up toward the end, thus furnishing a nearly uniform driving force for the pump piston, or plunger.

EXAMPLES OF FLYWHEEL PUMPING ENGINES

43. Fig. 24 shows a section of one side of a *Holley-Gaskill* compound pumping engine. The engine is double, the other side being like the one shown in the figure, the two engines having a common flywheel and crank-shaft, with cranks set 90° apart. The high-pressure cylinder *a* is placed directly over

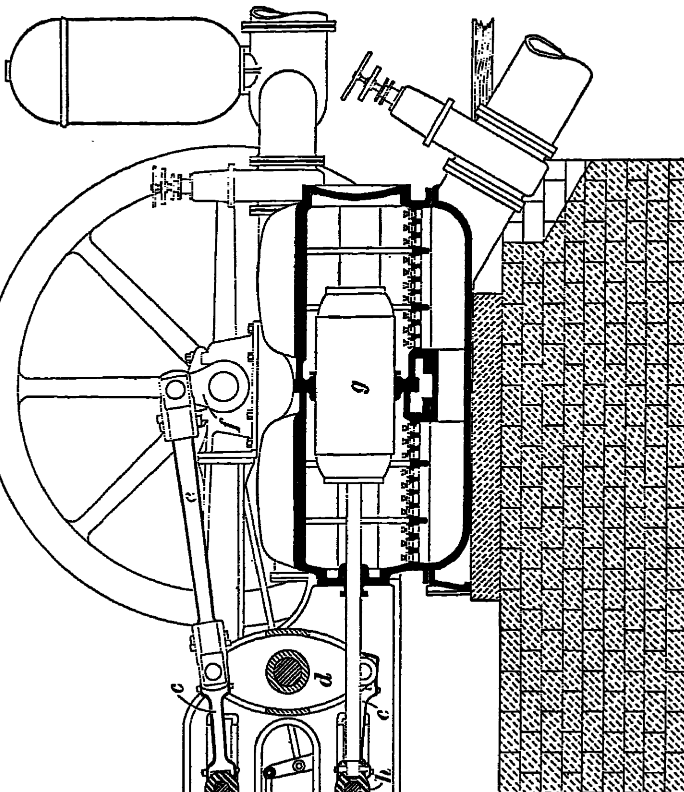


FIG. 24

the low-pressure cylinder *b* with short passages between them. The connecting-rods *c* from the two cylinders are attached to the opposite ends of a short walking beam *d*. By this arrangement the pistons move in opposite directions and the exhaust from the high-pressure cylinder passes directly to the low-pressure one. The valves are of the Corliss type, with a releasing gear for regulating the cut-off in the high-pressure cylinder. The connecting-rod *e* that actuates the crank *f* is attached to the upper end of the walking beam, and the rod that works the pump plunger *g* is attached to the crosshead *h* of the low-pressure piston.

44. Fig. 25 is a front and side elevation of a modern high-duty triple-expansion pumping engine erected at the North Point pumping station, Milwaukee, Wisconsin. The engine is of the vertical inverted three-cylinder type, having the pumps in line with the cylinders, and is condensing, the condenser not being shown. Each piston is connected to a separate outside-packed single-acting plunger by means of pump rods, as *a*. There are four pump rods to each plunger, which are joined to the steam crossheads *b* and straddle the crank-shaft *c* in such a way as to allow the cranks *d* to rotate freely between them. Two flywheels *e* are employed to give uniform rotation to the machine. In the figure, *f* is the suction pipe; *g* is the delivery pipe, the delivery from each chamber being connected to a common delivery main not shown in the illustration; *h* is the air chamber; at *i* are the suction valves; at *k* are the delivery valves; *l* are the plungers and *m* the pump chambers; *n* is one of the valve chambers, the upper part of which forms the delivery air chamber *h* and also supports the fronts of the bedplates. The rear of the bedplates is supported on the masonry foundation. The steam cylinders are provided with Corliss inlet and exhaust valves on the high and intermediate cylinders and Corliss inlet valves and poppet exhaust valves on the low-pressure cylinders. Large reheating receivers *o* are used between the high and intermediate cylinders and between the intermediate and low-pressure cylinders. An air pump *p* is driven directly from the plunger crossheads and serves to

remove the water of condensation, etc. from the condensers. An air-charging pump *q* pumps a small quantity of air into the water in order to replenish the air supply in the air chambers. A jacket drain pump *r* drains the water from the steam jackets. A suction air chamber *s* is fitted to the extreme end of the suction pipe and prevents shocks.

Pumps of the design shown in Fig. 25 are used almost exclusively for high-duty municipal waterworks service and are extremely economical.

45. Fig. 26 shows another type of high-duty municipal pumping engine, view (*a*) being a side elevation and view (*b*) the end elevation. This pump is of the crank-and-flywheel type; the motion of the pistons is not converted into a rotary motion in the manner shown in Fig. 25, but through the intervention of a rocking beam *a*, which is rocked back and forth by the high-pressure piston and low-pressure piston and is connected to the crank and flywheel by the connecting-rod *b*. This design, from its designer, is known as the **Leavitt** design. Pumps of this type have rather more parts than the type shown in Fig. 25, but they are not so high and are more accessible. The pumps are of the plunger type and are inside-packed; in the illustration, *c* indicates the plungers, *d* the pump chambers, and *f* the inside plunger packings. The tops of the pump chambers form delivery air chambers. The suction valves are located at *h* and the delivery valves are at *i*; the delivery pipe *j* discharges the water through the surface condenser *k*, thus using the delivery water for condensation. A butterfly valve *l* controls the amount of water passing through the condenser *k*. The exhaust pipe *m* from the low-pressure cylinder enters the top of the condenser; the pipe *n* leads from the condenser to the air pump *o*. This pump is double-acting and is driven from an arm attached to the beam *a*. Two reheating receivers *p* are used to heat the steam from the high-pressure cylinder during its passage to the low-pressure cylinder. The lower ends of the pump chambers rest directly on the bottom of the pump well, which is open to the river from which the pump takes its water. The water inlets are at *q* all

around the base of the pump. This pump is located at Louisville, Kentucky.

46. Small flywheel pattern steam pumps are made by many pump manufacturers for service where economy of operation is considered more desirable than low first cost. An example of a small pump of the flywheel pattern is the **Sewell-Cameron** steam pump shown in Fig. 27, built by the A. S.

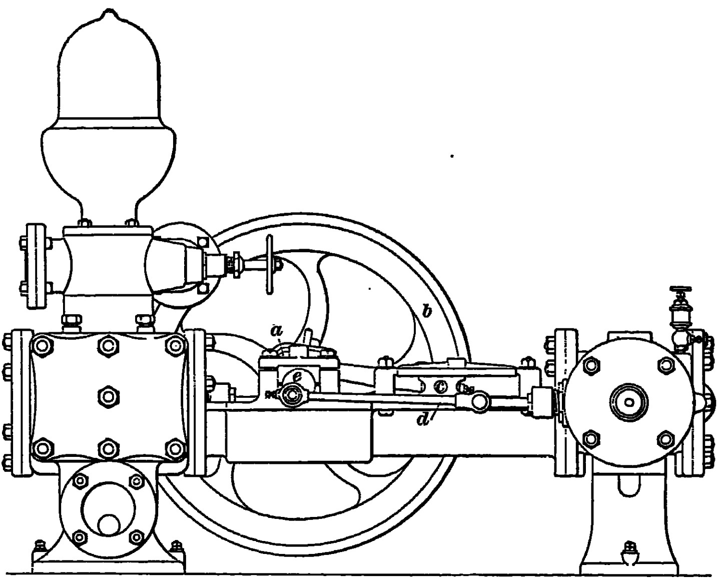


FIG. 27

Cameron Steam Pump Works, New York City, New York. The steam end of the pump is a regular plain slide valve steam engine, the crank-shaft *a* of which carries the flywheel *b* at one side and a small crank *e* at the other side. The crank *e* drives the slide valve through the valve rod *d*. The crosshead *c* is connected to the crank-shaft *a* by a connecting-rod, as shown, and is also connected to the piston rod of the water end, so that the steam piston and water piston move together.

WATER ENDS OF RECIPROCATING PUMPS

TYPES OF WATER ENDS

47. Reciprocating pumps, according to the construction of the water end, are divided into two general classes, which are *single-acting* and *double-acting pumps.*

Single-acting pumps may be either of the bucket pattern or of the plunger pattern. The packing that makes the plunger water-tight is always on the outside of the pump, and either one or two plungers may be employed; in the latter case, the two plungers are in line with each other and joined by yokes and rods outside the pump.

Single-acting lifting pumps are rarely employed except for pumping from deep wells, to which work they are peculiarly well adapted on account of the very long rod driving the pump bucket being under tension while the liquid handled is being lifted. A single-acting plunger pump is not adapted for this class of work, because, in the first place, it requires a stuffing-box, which is inaccessible without drawing the whole pump from the well, and in the second place, the rod driving the plunger is under compression while lifting the liquid, and hence will buckle, owing to its great length.

Double-acting pumps are made both in the piston and plunger patterns; piston pumps are naturally inside-packed, but plunger pumps of this class may be either inside-packed or outside-packed. The great advantage of outside packing is its accessibility; when pumping against high pressures, the packing requires frequent renewal, and hence the outside-packed pump has become of late years a favorite for this service. Another great advantage is that leaks are instantly observed.

CONSTRUCTION OF TYPICAL WATER ENDS

48. The construction of the water end of a piston pattern double-acting pump has been shown in Fig. 2. The piston is usually made a water-tight fit in the pump cylinder by means

of rings of fibrous packing placed around the circumference of the piston. It will be apparent that water may leak past the water piston, owing to wear of the packing, to a considerable extent without being discovered, the result of this being that then the pump discharges less than its normal quantity of the liquid handled. Owing to this piston pumps are illy adapted for working against very high pressures.

49. The principle of construction of a double-plunger single-acting pump is shown diagrammatically in Fig. 28. The pump barrel *a*, cylindrical in form, has stuffingboxes *b* at both ends, through which pass the two plungers *c* and *d*. A division plate *e* in the middle of the barrel separates it into two parts. The two plungers are united outside the pump barrel by the

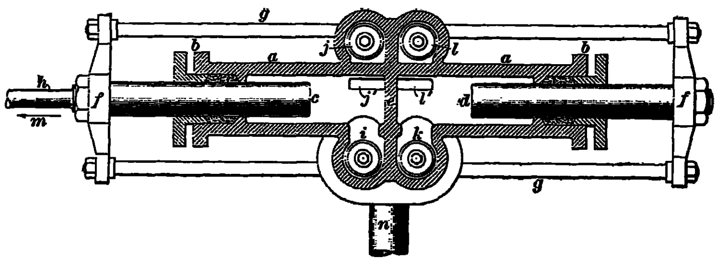


FIG. 28

yokes *f* and tie-rods *g*; the piston rod *h* leads to the steam cylinder. The left-hand side of the pump is supplied with the suction valve *i* and delivery valve *j*; the suction valve *k* and delivery valve *l* serve the right-hand side of the pump.

Assume that the piston rod *h*, and hence the plungers *c*, *d*, is moving in the direction of the arrow *m*. Then, the suction valve *i* is open, the water flowing in from the suction pipe *n*, and the delivery valve *j* is closed. While the plunger *c* is filling its pump barrel, the plunger *d* is displacing water, its suction valve *k* being closed, and its delivery valve *l* being open, the water flowing below the delivery valve through the passage *l'* and past the delivery valve into the delivery pipe. The passage *j'* leads below the delivery valve *j*.

From the explanation just given, it will be apparent that this type of pump consists simply of two single-plunger pumps

placed end to end, so that when the one plunger is lifting water the other one is discharging. In other words, a double-plunger pump of the type shown will give a delivery equivalent to that of a single double-acting pump of the same diameter and stroke.

The water end of a double-plunger single-acting pump is often made duplex, that is, two such water ends are placed side by side; this type of pump, being outside packed, is well adapted to pumping against very high pressures.

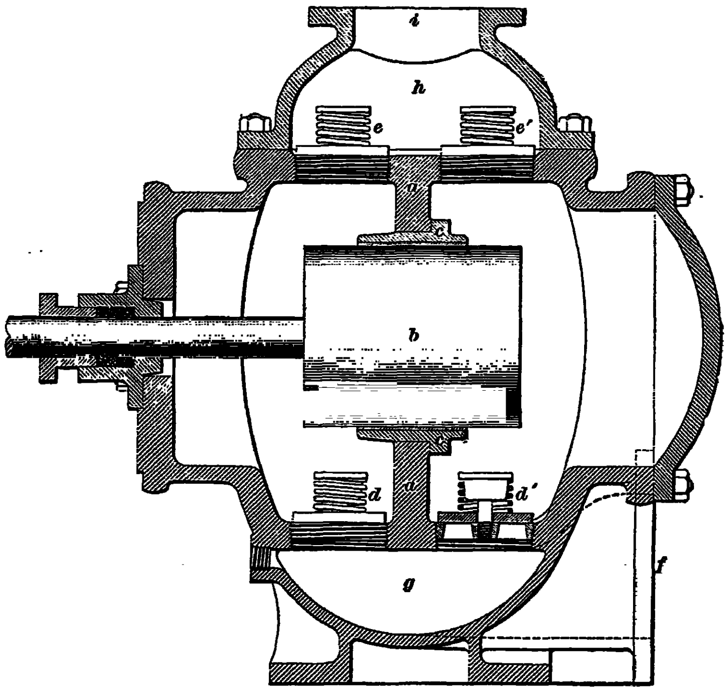


FIG. 29

50. Fig. 29 shows the water end of a double-acting inside-packed plunger pump. The pump chamber is divided into two parts by a partition *a*, through which the plunger *b* works back and forth. A water-tight joint between the plunger and partition is made either by a closely fitting bronze-lined bushing *c* or a regular stuffingbox and gland and fibrous packing. On either side of the partition is a set of suction valves *d*, *d′* and

delivery valves e, e'. The water enters the pump through the suction pipe, which is connected at f, and flows into the suction-

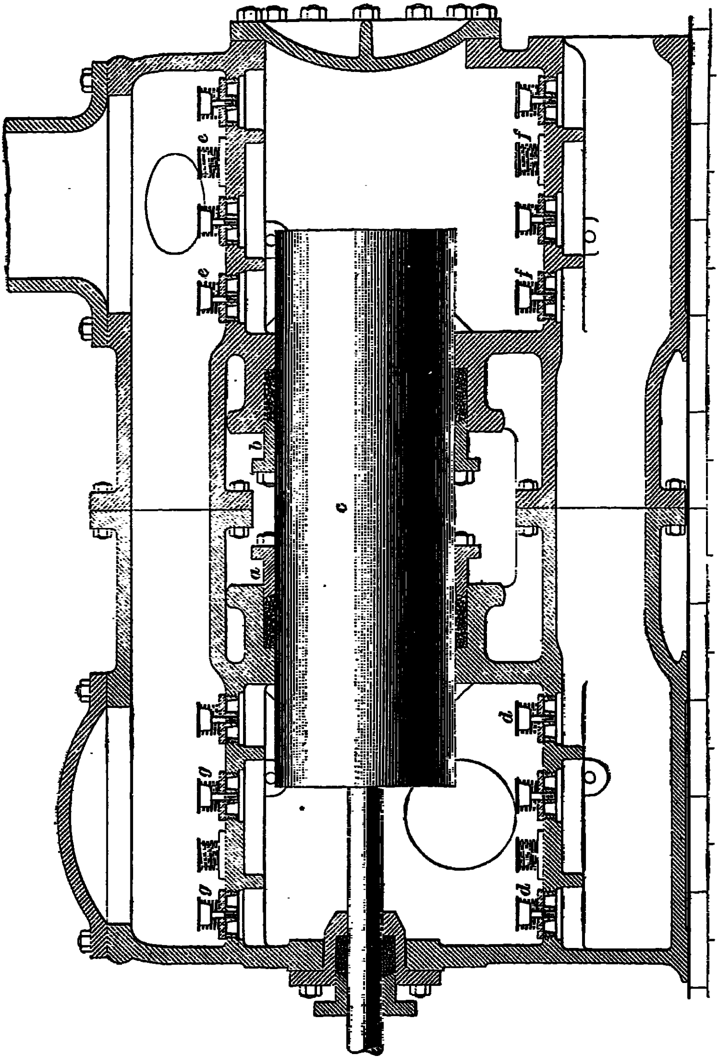


FIG. 30

valve chamber g, whence it passes to either side of the partition a and then into the delivery-valve chamber h and into the

delivery pipe connected at *i*. When the plunger moves to the right, it displaces the water on the right of the partition *a*; the suction valve *d'* is closed by the pressure existing there, while the delivery valve *e'* is open and the water discharges into *h*. At the same time, the plunger creates a partial vacuum at the left of the partition *a* and, hence, water flows through the open suction valve *d* into the left pump chamber. The delivery valve *e* is kept closed by the pressure in *h*. When the plunger moves to the left, the suction valve *d'* and delivery valve *e* open and the suction valve *d* and delivery valve *e'* close. It is thus seen that the pump discharges during either stroke of the plunger, that is, the pump is double-acting.

51. Fig. 30 shows a sectional view of the water end of a center-packed double-acting plunger pump, the stuffingboxes *a* and *b* being used for packing the plunger *c*. The action of the pump is identical with that of the pump shown in Fig. 29, that is, when the plunger moves to the right the suction valves *d, d* and delivery valves *e, e* are open and the suction valves *f, f* and delivery valves *g, g* are closed. When the plunger moves to the left, the suction valves *f, f* and delivery valves *g, g* are open and the suction valves *d, d* and delivery valves *e, e* are closed.

52. Fig. 31 shows, in diagrammatic form, two forms of a plunger pump that is double-acting and is known as a **differential pump.** Its distinguishing feature is that it needs only one set of suction valves and delivery valves. Fig. 31 (*a*) shows the arrangement used for two plungers *a* and *b*, which are connected together by yokes and side rods. In Fig. 31 (*b*) the two plungers are connected directly together. In both designs one plunger, as *a*, has exactly double the area of the other plunger *b*; this fact must be carefully borne in mind. Since the stroke of both plungers is the same, it follows that the larger plunger in Fig. 31 (*a*) will displace double the quantity of water that the smaller plunger displaces. In Fig. 31 (*b*) the left-hand side of the plunger *a* displaces double the quantity of water displaced by the right-hand side of the plunger. In both designs *c* is the suction valve and *d* the delivery valve.

The operation of the differential pump shown in Fig. 31 is as follows: The pump being filled with water and the plungers moving to the right, the suction valve is open and the delivery

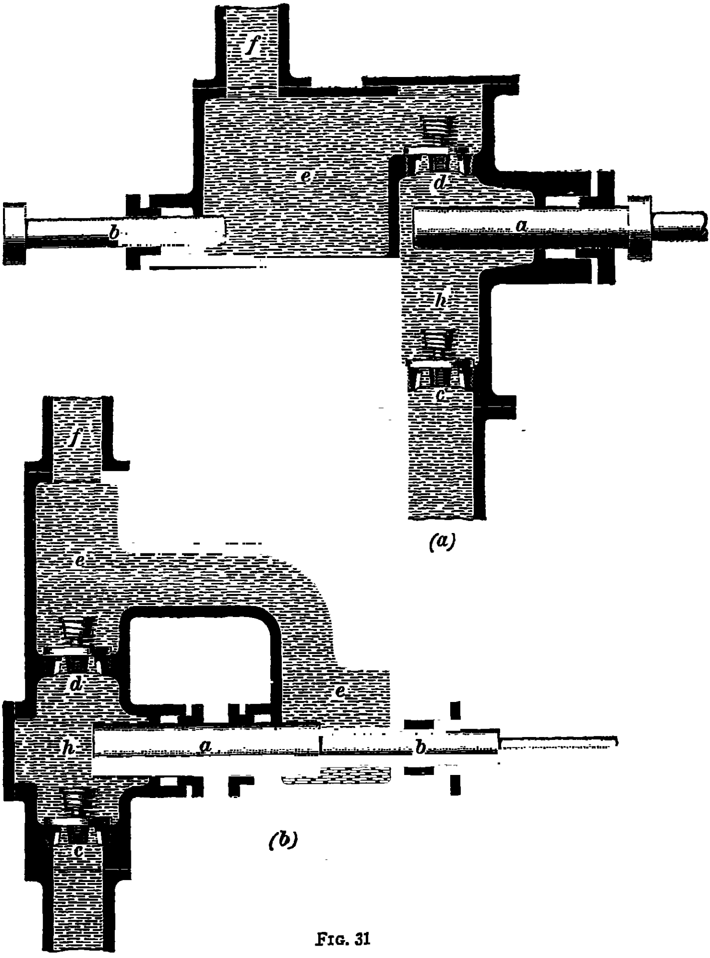


Fig. 31

valve closed. The plunger *b*, or the right-hand side of the plunger *a*, Fig. 31 (*b*), forces a volume of water equal to its displacement out of the chamber *e* and up the delivery pipe *f*. At the same time, double the volume of water is drawn into

the suction chamber *h*. Now, assume that the plungers move to the left. The suction valve is then closed and the delivery valve is open, and double the quantity of water discharged during the stroke to the right now flows into the chamber *e*. But while this is going on, the volume of the chamber *e* increases by the receding of the plunger *b*, or the outward movement of the plunger *a* in Fig. 31 (*b*), by an amount that at the end of the stroke is equal to exactly one-half the amount discharged into it, so that the outflow into the delivery pipe is only one-half of that discharged into the chamber *e*. This outflow is equal to the displacement of the small plunger, or the right-hand end of the plunger *a* in Fig. 31 (*b*), and hence the same amount of water is discharged during both strokes.

PUMPS

(PART 1B)

POWER AND DISPLACEMENT PUMPS

POWER PUMPS

PURPOSE OF POWER PUMPS

53. Pumps in which the piston or plunger is driven by a crank, an eccentric, or a cam that receives its motion through a belt or gearing from some outside source of power are usually called **power pumps.**

A single power pump is one in which but one pump is driven by the shaft. This pump may be either *single-acting* or *double-acting.*

When two pumps are driven by cranks, eccentrics, or cams on a single shaft, the combination is called a **duplex power pump.** The discharge branches from the two pumps are generally combined in such a way that they discharge through a single pipe; and by a proper arrangement of the driving members, the flow through the discharge pipe and the power required to drive the pumps are made nearly constant. If the pumps are single-acting and the driving members are set 180° apart, the discharge from the two pumps will be the same as the discharge from one double-acting pump with the same diameter of piston and length of stroke. Duplex double-acting pumps, with driving members set 90° apart, are much used and give a very steady discharge, since, when one driving

member is on its dead center and its piston, consequently, is at the end of its stroke and momentarily at rest, the other piston is moving at its maximum velocity and discharging at its maximum rate.

Three pumps driven by driving members on a single shaft form a **triplex pump.** The most common application consists in the use of three single-acting plunger pumps with driving members set 120° apart. With such a combination, at least one of the pumps is always discharging and one taking water from the suction pipe, and the flow is therefore continuous and nearly uniform.

54. Where the supply of power is steady, a belt-driven power pump is very convenient and economical for the purposes for which such pumps can be used, since they get their power with the same degree of economy as the engine by which they are driven; they are also simple in construction and easily operated.

In locations where there is no steam available, or where the use of the pump is so intermittent that a steam plant will not be economical, or where the cost of supplying steam is too great, but electric current is available, power pumps driven by electric motors may be used to advantage.

Small pumps driven by windmills, hot-air engines, gas engines, etc. are much used for supplying water to buildings that have no connection with public waterworks. Small single-acting plunger pumps are most commonly used with these methods of driving, although double-acting pumps are sometimes used.

Where water-power is available, pumps for city waterworks or for supplying manufacturing establishments are often driven by waterwheels or hydraulic turbines.

EXAMPLES OF POWER PUMPS

55. A **Davis** duplex power pump of the vertical type, made by J. B. Davis & Sons, Hartford, Connecticut, arranged to be driven by belt from any convenient source of power,

is shown in Fig. 32. The pump illustrated has two plungers *a* which are driven from the countershaft *b* by eccentrics, as *c*, which are the mechanical equivalent of a crank in this case. The countershaft is driven from the main shaft *d* by the gears *e* and *f*, which insure a low speed of the plungers in order to obtain quiet action of the pump. The main shaft *d* carries a tight pulley *g* and a loose pulley *h*, which permits starting and stopping the pump by shifting the driving belt

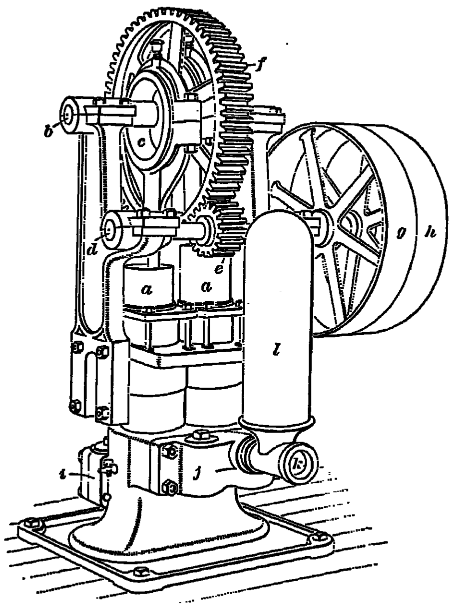


FIG. 32

from one pulley to the other without affecting the source of power. The suction inlet cannot be seen in the illustration; the suction valves for the two plungers are contained in the suction valve chest *i* and the delivery valves in the delivery valve chest *j*. The delivery pipe attaches at *k*; a delivery air chamber *l* promotes a steady delivery of water by the pump.

56. Fig. 33 shows a type of **triplex belt-driven power pump** much used for feeding boilers, filling elevated tanks in buildings, supplying hydraulic elevators, etc., and built by The

Goulds Manufacturing Company, Seneca Falls, New York. It consists of three single-acting plunger pumps *a* driven by cranks *b* set at 120° on a single shaft *c*. A loose pulley *d* and a tight pulley *d'* provide means for starting and stopping the pump without disturbing the engine or main shaft. The pulley shaft *e* is geared to the crank-shaft *c* by a pinion (not shown)

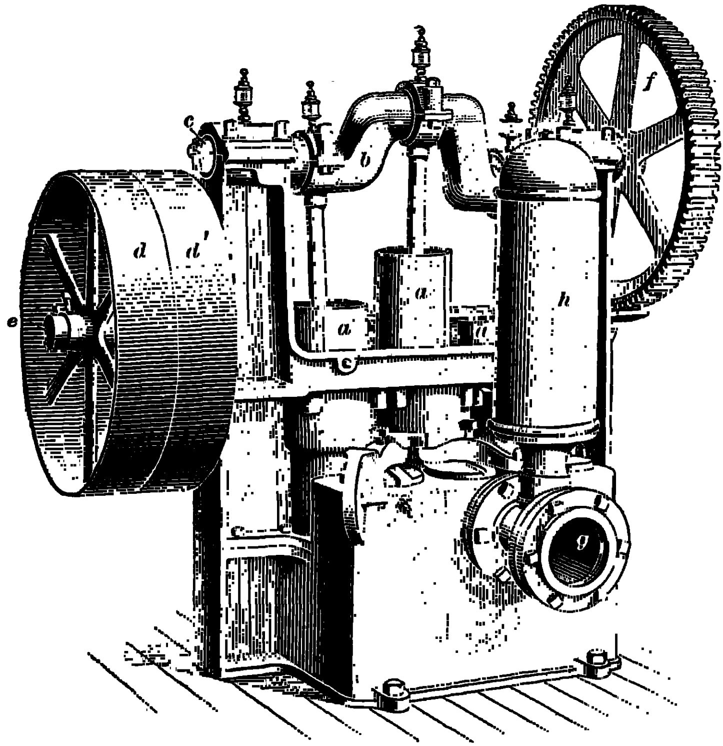


FIG. 33

and spur wheel *f*. The suction inlet of the pump is not shown, being on the side opposite the discharge opening *g* and the air chamber *h*.

57. Instead of driving a power pump by a belt, it may be driven by an electric motor, as shown in Fig. 34, which is coupled directly to the main shaft of the pump; this is built by the same company as that shown in Fig. 33. As electric motors

usually run at high speed, a triple gear reduction is used in this case. It will be obvious that a steam engine or other source of power may be coupled directly to the pump.

58. A vertical section of a power pump of the **single-acting plunger type** is shown in Fig. 35 (*a*). The pump is driven by a belt running on the pulley *a*, from which motion is transmitted to the plunger through the pinion *b*, gear *c*, shaft *d*, and connecting-rod *e*. A crosshead *f* sliding in guides *g* prevents side thrusts on the plunger, and a stuffingbox *h* prevents

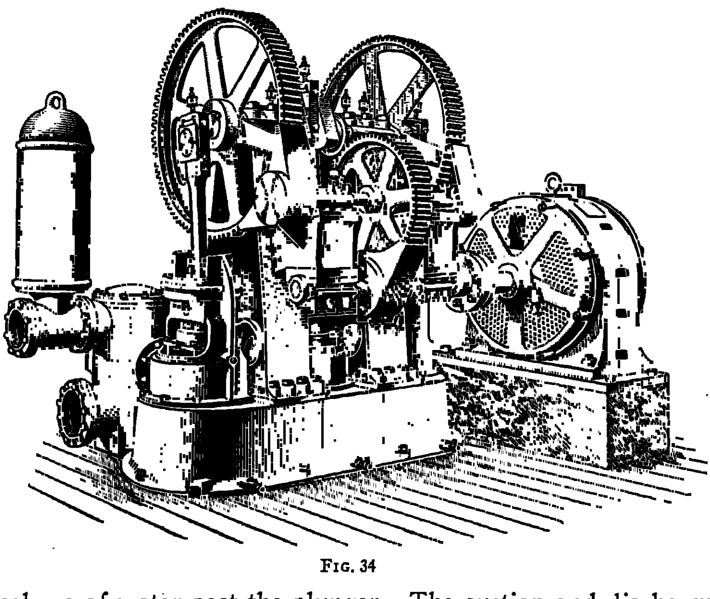


FIG. 34

leakage of water past the plunger. The suction and discharge valves are contained in the casing *i*, the suction and discharge pipes being connected at *j* and *k*, respectively.

59. Fig. 35 (*b*) shows a vertical section of the water end of a **double-acting piston pump.** The piston rod *a* is given a reciprocating motion by means of a crank and connecting-rod, as shown in Fig. 35 (*a*). However, there are two sets of suction and discharge valves, since the pump is double-acting. The suction valves *b* and *b′* communicate with the

suction pipe *c*, and the discharge valves *d* and *d'*, with the discharge pipe attached at *e*. All the valves are contained in the

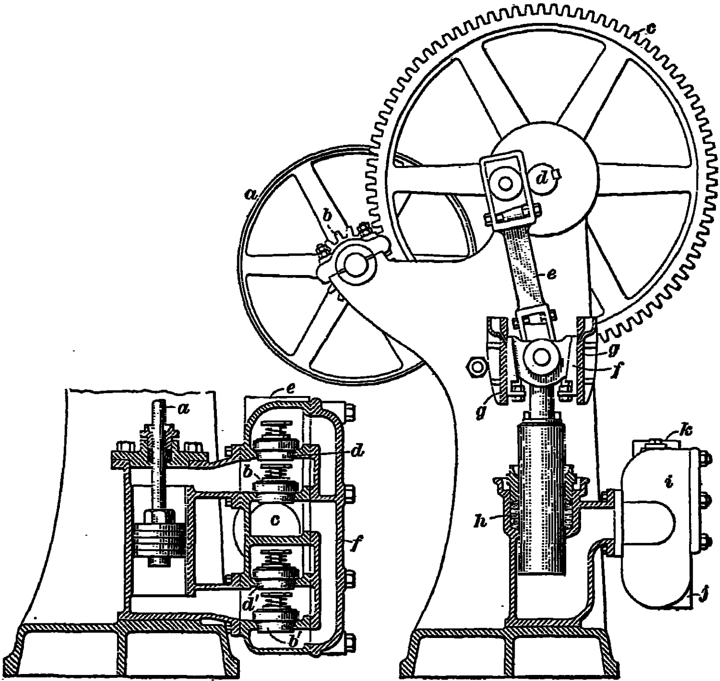


FIG. 35

casing *f*, which is bolted firmly to the pump frame. The construction of the valves in Fig. 35 (*a*) is similar to that shown in Fig. 35 (*b*).

DISPLACEMENT PUMPS

DEFINITION

60. A **displacement pump** is a piece of apparatus in which there is a complete absence of moving parts, such as plungers, buckets, or pistons, and where the liquid to be pumped is moved by some fluid under pressure, such as live steam, compressed air, or a burning mixture of air and gas. Of the

steam-operated displacement pumps, the oldest and most widely known, but not the only one, is the pulsometer. The best known compressed-air-operated displacement pumps are the Harris compressed-air direct-air-pressure pump and the Pohlé air lift. A comparatively recent example of a displacement pump operated by the pressure generated by burning a mixture of air and gas is the Humphrey gas pump.

PULSOMETER

61. Fig. 36 presents a perspective view and Fig. 37 a sectional view of the latest type of pulsometer. There are two water chambers *a* and *a′* and between them a vacuum chamber *b*, which is not in direct communication with them but is connected with the suction chamber *c*. The steam pipe is attached to the neck piece *d*; the suction pipe is attached at *e*, and the discharge pipe at *f*. There are four oval handholes covered by plates *g*, which are arranged in places suitable for access to the valves and only three of which can be seen in Fig. 36 and two in Fig. 37. A hook *h* permits the ready attachment of tackle in placing the pulsometer in position or moving it. The flange *i* is put on the foundation.

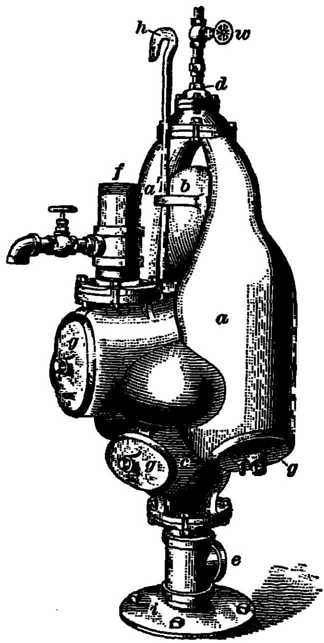


FIG. 36

The sectional view shown in Fig. 37 is taken so as to pass through the water chambers *a*, *a′*, the vacuum chamber *b*, and the suction chamber *c*. Just below the neck piece *d* is the ball valve *j*. The foot-valve *k* is a flap valve and is just above the suction-pipe connection *e*; the discharge pipe *f* is shown by dotted lines. The handhole plates *g* are held in place by a yoke *l* and a bolt and nut *m*. The suction

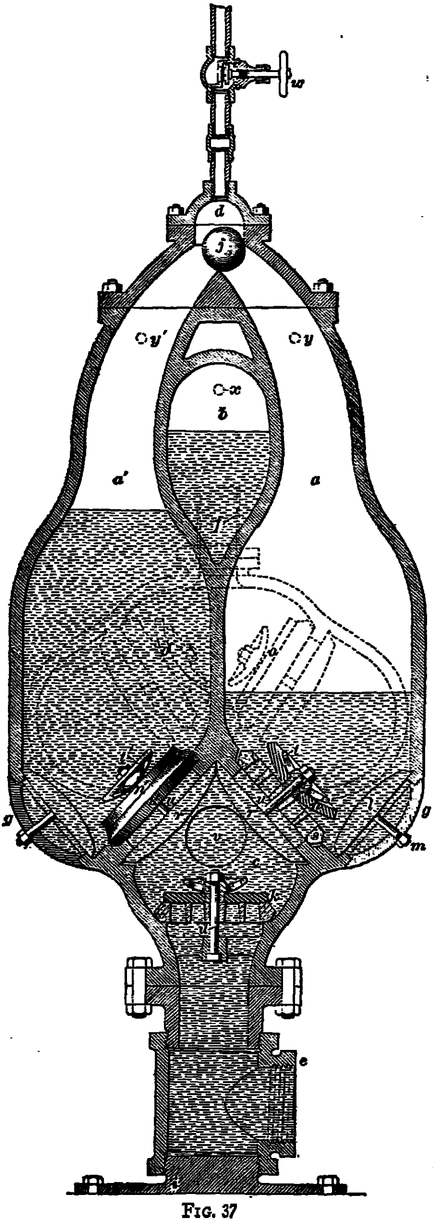


FIG. 37

valves n, n' and the discharge valves o, o' are flap valves, made so that the seats s, s', clamps r, r', and valve guards t, t' are readily removed and replaced. All the flap valves are like those shown in section at k and n. They are held in place by a nut and bolt u. The foot-valve k holds the water in the pump and opens upwards when a new charge is drawn into one of the water chambers a or a'. The suction valves n or n' open whenever water is drawn into the water chambers a and a'. The discharge chamber, shown by dotted lines in Fig. 37, communicates with the water chambers a and a' and contains the discharge valves o and o'. The suction chamber c forms an indirect communication between the water chambers a and a' and the vacuum chamber b, through the opening v.

62. The action of the pulsometer is as follows: Both water

chambers *a* and *a′* are filled with water to about the height of the water shown in *a′*, Fig. 37. The steam valve *w*, Fig. 36, is then opened and the steam enters one of the two chambers, as, for example, *a′*, Fig. 37, the ball valve *j* then being at the right, as shown. The water in *a′* will be forced through the discharge valve *o′* into and up the discharge pipe *f* and will continue to flow until the water level in *a′* falls below the edge of the discharge opening. At this point, the steam and water mix in the discharge passage and the steam is condensed, creating a vacuum in *a′*. The pressure in *a* being now greater than in *a′*, owing to this vacuum, the ball *j* is shifted to the left, permitting the steam to enter the chamber *a*. The water in this chamber is now driven out through *o* into the discharge pipe *f* until the steam mixes with the water in the discharge passage and a vacuum is formed in *a*, as just described. While this is being done, however, the pressure of the atmosphere, the moment the vacuum is formed in *a′*, forces the water up the suction pipe *e* and through the suction valves *k* and *n′* into the chamber *a′*, again filling it with water and destroying the vacuum there. When the suction valve of either chamber closes, owing to the shifting of the ball *j*, the water from the suction pipe enters the vacuum chamber *b* through the connection *v*, and is brought gradually to rest by the compression of the air in *b*, thus preventing a shock due to a sudden stoppage of the inflowing water. When the water in *a* has reached the level shown, the steam in *a* is condensed, the ball *j* is shifted to the right, and *a′* becomes the driving chamber.

63. A small air valve *x*, Fig. 37, admits air to the vacuum chamber *b*, to replenish that which is lost through leakage and through absorption by the water; two similar valves *y* and *y′* admit a small quantity of air to the chambers *a* and *a′*, respectively, just before the suction begins. This interferes with the suction somewhat, but is necessary in order to regulate and govern the amount of water admitted to the chambers, and to prevent the steam from condensing before the water gets below the edge of the discharge outlet. If there is a partial vacuum formed in *a*, owing to condensation of the steam, the atmos-

pheric pressure opens the valve *y* and admits a little air to the chamber. The incoming water compresses this air and soon closes the valve. When the air has been compressed to such an extent as to balance the outside pressure of the atmosphere, the suction valve *n* will close and no more water can get in. Since the same thing occurs in the other chamber *a'*, it is evident that the amount of air admitted controls the amount of water admitted during the suction period, more water entering when there is less air in the chamber and vice versa. The admission of the air can be so adjusted that the suction valve in either chamber will close at the instant the ball is shifted to the other side to admit steam.

The cushion of air between the steam and the water prevents them from coming in contact during the forcing process until the water level has sunk below the edge of the discharge orifice. Air being a poor conductor of heat, the steam does not condense until the mixture of the steam and water has taken place.

64. The pulsometer will raise water by suction to about 75 per cent. of the theoretical height corresponding to the existing barometric pressure. Thus, at sea level, with the barometer standing at 30 inches, it will draw water about 26 feet ($34 \times .75 = 25.5$). It will force water, however, to a height of 100 feet. The pulsometer has no wearing parts whatever except the valves, which are easily and cheaply repaired. It will work in almost any position, and when once started requires no further attention. There are no parts that can get out of order. It will pump anything, including mud, gravel, etc., that will pass through the valves. Its first cost is low and there is no exhaust steam to dispose of and no noise. It uses about twice as much steam as an ordinary direct-acting steam pump of equal capacity.

65. The manufacturers of the pulsometer give the following directions for operating it:

"When the pulsometer has been placed in the desired position, the necessary suction, discharge, and steam-pipe connections have been made, and the air valves screwed into place, prime the pump by pouring water through the plugged opening

in the middle chamber provided for the purpose; when it is filled, quickly replace the plug. The water covering the valves insures their being air-tight. Close the three air valves tight, thus preventing the admission of air to the pump; also, close the steam-controlling globe valve at the pump, which is now ready to start. Open wide the valve at the boiler, then quickly open the steam-controlling globe valve at the pump a full turn, leaving it open 3 or 4 seconds, then close it quickly and allow it to remain closed about 4 or 5 seconds. Repeat this manipulation a few times until the steam valve ball in the neck of the pump clicks and rattles, indicating that the pump has caught its suction; then leave the steam turned on about one-half or three-quarters of a turn, and regulate the admission of air by opening the two side air valves about one-half of a turn, and then the middle one little by little until a regular stroke of the steam valve ball in the neck of pump is obtained.

"The admission of air is regulated as the depth of suction, height of delivery, and pressure of steam varies; therefore, no stated rules for the adjustment of the air valves can be given. With a few experiments, their proper management will readily suggest itself. Too free an admission of air is detrimental to the economical operation of the pulsometer, for it will fill the space that should be occupied by water, as each volume of water or air expelled from the chamber requires an equal volume of steam to perform the work. When too much air is admitted, quickening of pulsation and a reduction of the volume of outflowing liquid will result. When the admission of air is deficient, irregular pulsations will follow.

"On short suction, more air is required than on long suction, and the shorter the suction the greater the quantity of air required. On long suction, the air valve in the middle chamber may be entirely closed, and as little air admitted through the side air valves as will produce a regular stroke of the steam valve ball.

"When the pump has been started and the air valves have been adjusted, open further the steam-controlling globe valve at the pump little by little, waiting meanwhile to notice the increase in the number of strokes of the steam valve ball, until

no further increase can be obtained. This shows that the extent to which the valve should be opened has been reached, and any additional quantity of steam admitted by further opening the valve will be detrimental to the proper and economical working of the pump, as more steam will then be admitted to the chambers than can be condensed after the liquid is expelled; this will prevent the formation of a sufficient vacuum to permit of the chambers being entirely refilled, and will result not only in a waste of steam, but in causing the pump to work at less than its full capacity.

"When the globe valve has been set so that the proper amount of steam is being admitted, attention should be turned to the air valves, which may possibly require further slight adjustment. With the steam and air supply properly adjusted, the pulsometer may, if desired, be operated entirely by the valve at the boiler or main steam pipe, and will instantly start when steam is turned on, unless prevented by one of the following interruptions, which should at once be attended to: Strainer or suction pipe becoming choked; leakage in suction connections, permitting admission of air; some obstruction too large to pass the foot-valve; steam valve ball in neck piece becoming obstructed; foaming of boiler, whereby hot water is temporarily supplied instead of steam; steam pressure lower than required for the service, or starting on low steam pressure with the globe valve near the pump opened to suit the service, and then receiving a large increase of steam."

HARRIS DIRECT-AIR-PRESSURE PUMP

66. The direct-air-pressure pump here shown is the design of Professor Elmo G. Harris and is one of the simplest forms of pump. The pump is shown in diagrammatic form in Fig. 38. There are two pump tanks *a* and *b*, which are fitted with suction valves *c* and *d* and discharge valves *e* and *f*. The two tanks are connected to the common suction pipe *g* and both discharge into the same discharge pipe *h*. The tops of the pump tanks are connected by pipes *i* and *k* to an air compressor *m*, and by means of an automatically operated four-

way cock l, either tank can be connected alternately to the suction side and the compressor side of the air compressor.

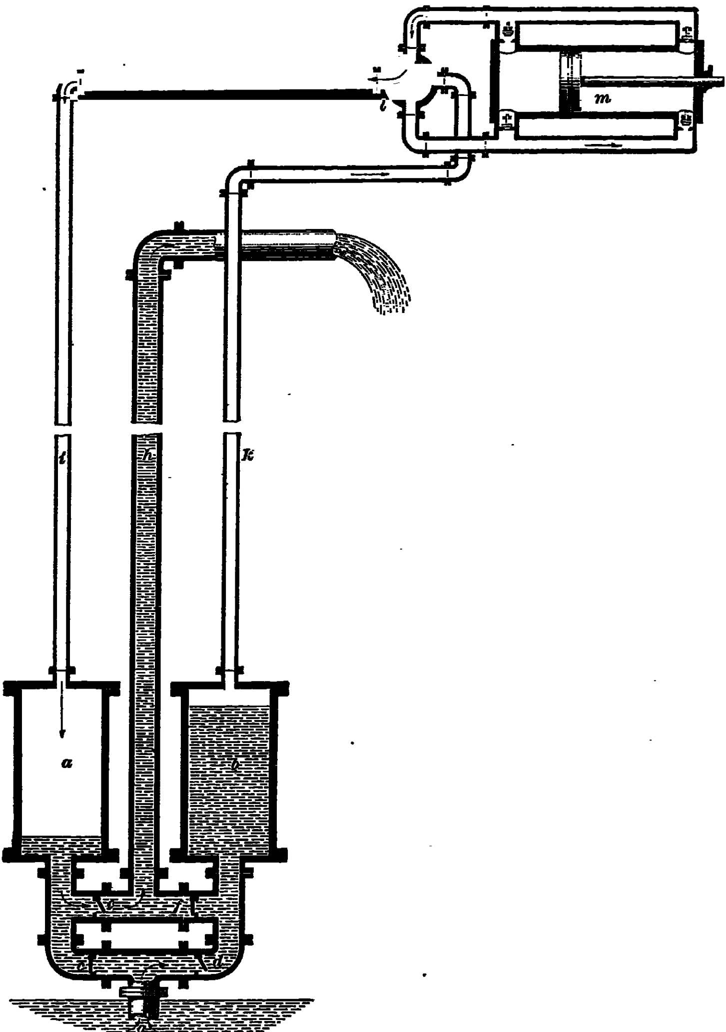


FIG. 38

The operation is as follows: with the cock l in the position shown, the tank b is connected to the suction side of the air

compressor, and hence a vacuum is formed in the tank *b*. Consequently, the water in the supply is forced by atmospheric pressure up the suction pipe *g*, lifts the valve *d*, and passes into the tank *b*. At the same time the tank *a* is connected to the compressor side, and the air pressure on top of the water forces it out, the water holding the suction valve *c* closed but opening the delivery valve *e* and passing up the discharge pipe *h*. When the tank *a* is nearly empty, the tank *b* is nearly full; the cock *l* is then turned automatically so as to bring the tank *a* in communication with the suction side of the air compressor and the tank *b* in communication with the compressor side. The water now flows into *a* and out of *b*, and the cycle of operations is repeated as long as the air compressor is working. The height to which water can be forced obviously depends on the pressure to which the air is compressed.

POHLÉ AIR LIFT

67. The **Pohlé air lift** is much used for pumping water from artesian wells; it is operated by means of compressed air and has no moving parts. It is not affected by sand or grit and the water is benefited to a considerable extent by the action of the air, in that it purifies and cools the water while it is being pumped. Another advantage claimed for this device is that it increases the yield of an artesian well from two to five times; also, the full area of the well is available for a flow of water. Compressed air is supplied by means of an air compressor at the surface, which may be located in any convenient position, or one air compressor may supply several artesian wells.

68. The operation of the pump is as follows: Two properly proportioned pipes are inserted in the well, using either of the three arrangements shown in Fig. 39. Compressed air is supplied through the pipe *a* to the bottom of the well tube *b*. At the beginning of the operation the water inside and outside of the pipe is at the same level. When air is forced in through the pipe *a*, it forms alternate layers with the water, so that the pressure per square inch of the column thus

made up of air and water inside the water pipe is less than the pressure per square inch outside the pipe. This difference of pressure causes a continuous flow from the outside to the inside of the water pipe, and its ascent is constant and is free from shock or noise of any kind. The strata of compressed air in

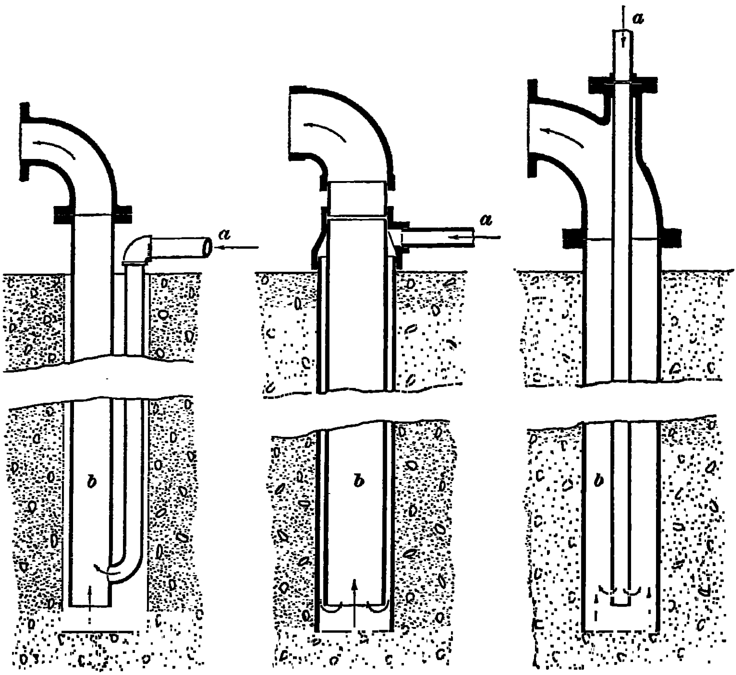


FIG. 39

their ascent prevent any slipping back of water. As each stratum progresses upwards to the spout, it expands on its way in proportion to the overlying weight of water, so that the pressure of the air gradually becomes less and finally reaches the atmospheric pressure.

HUMPHREY GAS PUMP

69. The Humphrey gas pump, made in the United States by The Humphrey Gas Pump Company, Syracuse, New York, differs from other displacement pumps in that the pressure

required to force the water in the pump to a higher level is created by the burning of a combustible mixture of air and gas, which has been compressed by the action of the water itself in the pump chamber.

A Humphrey pump is shown in diagrammatic form in Fig. 40, in order that its principle of operation may be readily explained. The pump consists of a vertical cylinder *a* closed at the top; its lower end has a large number of inwardly opening water valves *b* and is in communication by a pipe *c* with the source of water supply *d*. The lower end of the pump cylinder also connects to the discharge pipe *e*, which is sometimes called the *play pipe*, and which opens into the reservoir *f* into

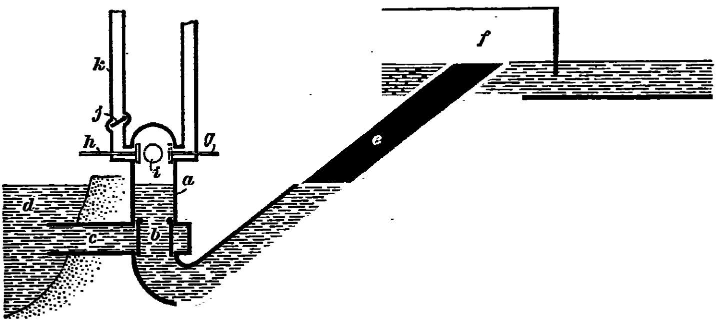


FIG. 40

which the water is to be discharged. The upper end of the pump cylinder carries three sets of inwardly opening valves, only one of each set being here shown; the valve *g* is an inlet valve by means of which a fresh mixture of gas and air is admitted to the pump cylinder. The valve *h* is an exhaust valve by means of which a large part of the products of combustion of the burned mixture of gas and air is allowed to escape from the pump cylinder; the valve *i* is a scavenging valve by means of which, after the mixture of gas and air has been burned, fresh air is permitted to enter the pump cylinder before exhaust begins, in order to dilute as much as possible the products of combustion. An inwardly closing check-valve *j* in the exhaust pipe *k* prevents exhaust gases from returning to the pump.

70. The operation of the pump is as follows: With the pump at rest, the water stands at the same level in the source of water supply, the pump cylinder, and the play pipe. The valves *b, g, h, i,* and *j* are now in their closed position; to start the pump a mixture of gas and air is pumped into the space of the pump cylinder *a* above the water level in it and is compressed by the same separate compressor that forces the mixture in; the combustible mixture is then ignited by an electric spark and in burning creates a high pressure. This pressure sets the water in the pump cylinder *a* and the play pipe *e* in motion, forcing it upwards in the play pipe *e,* the expansion of the burning mixture accelerating the motion of the water. Owing to the kinetic energy of the moving body of water this continues in motion after the burning mixture in the upper part of the pump cylinder has expanded below the pressure of the atmosphere; owing to the partial vacuum now existing in the pump cylinder the scavenging valves *i,* the exhaust valves *h,* and the water inlet valves *b* open, but the check-valve *j* remains closed. Air now rushes into the pump cylinder through the scavenging valve, as this communicates directly with the atmosphere, and dilutes the products of combustion in the upper part of the pump cylinder; at the same time water from the source of supply flows past the valves *b,* most of which inflowing water follows the moving water column in the play pipe *e,* while part of the inflowing water tends to raise the water level in the pump cylinder. As soon as the kinetic energy of the moving water column has expended itself by forcing water into the reservoir *f,* the water column with which the play pipe is filled comes to rest and then flows back towards the pump cylinder, the water valves of which close immediately. The scavenger valves *i* having been promptly closed by a spring as soon as the pressure in the pump cylinder exceeded the atmospheric pressure and before the beginning of the return flow of the water, and the exhaust valves *h* being held open, the returning water fills the pump cylinder, driving out the diluted products of combustion until the water reaches the exhaust valves *h* and closes them by its impact against them. The diluted products of combustion still remaining in the pump cylinder are

now compressed to a high pressure until the kinetic energy of the returning water column is completely expended, when the column of water comes to rest, and is then set in motion toward the reservoir *f* again by the pressure of the expanding compressed products of combustion. When the water level has fallen below that of the inlet valves *g*, the pressure in the upper part of the pump cylinder has dropped below that of the atmosphere, and consequently the mixture inlet valves *g* open, the exhaust valves *h* remain closed, and a combustible mixture is drawn into the pump cylinder until the kinetic energy of the outflowing water has once more been expended. The water column in the play pipe *c* now begins its second return movement and compresses the fresh mixture of air and gas in the pump cylinder, which mixture is ignited by an electric spark as soon as the returning water column comes to rest.

It will be understood that the compressor furnishing the initial compressed fresh charge of combustible mixture to start the pump, is closed down after the first explosion. Thereafter the pump works automatically, drawing in a fresh charge on the one forward movement of the water column, compressing it on the return movement, burning the fresh charge and finally diluting the burned charge with air on the next forward movement, and exhausting the burned charge on the next return movement.

71. The object of diluting the burned charge with air by means of the scavenger valves is to leave as small a quantity of the products of combustion as possible in the pump cylinder, because these products of combustion when mixed with the fresh charge reduce the pressure produced by the burning of the fresh charge in proportion to the quantity mixed with the fresh charge.

In an actual Humphrey pump the mixture inlet valves and the exhaust valves, and sometimes the scavenging valves as well, are mechanically interlocked. Thus, during the burning of the fresh charge the inlet valves and exhaust valves are both locked in their closed positions; the exhaust valves and scavenging valves are unlocked at the end of expansion, the exhaust

valves being forcibly pushed open by their springs as soon as they are unlocked. The scavenger valves, if of the interlocked type, are interlocked with the inlet valves so that both cannot open at the same time, and when unlocked are opened by the pressure of the atmosphere, and are closed by their springs. Directly after the exhaust valves have been closed by the impact of the water against them, they are locked in their closed position and the closed inlet valves are unlocked at the same time.

In a broad way the locking and unlocking of the valves is accomplished by a cylinder containing a movable piston and having one end connected to the play pipe; a rise in pressure in the play pipe pushes the piston outwards and a drop in pressure causes the piston to return under the action of a spring. The piston is connected to the locking levers of the valves and operates these levers so as to lock and unlock the valves at the proper time.

So many different forms of interlocking mechanism are used on Humphrey pumps that even brief descriptions of them are not feasible; however, when examining an actual pump of this type, the operation of the particular interlocking mechanism employed is readily disclosed by a careful inspection. The details of construction of Humphrey pumps also vary considerably, and for this reason only the broad principles underlying their action have been given here.

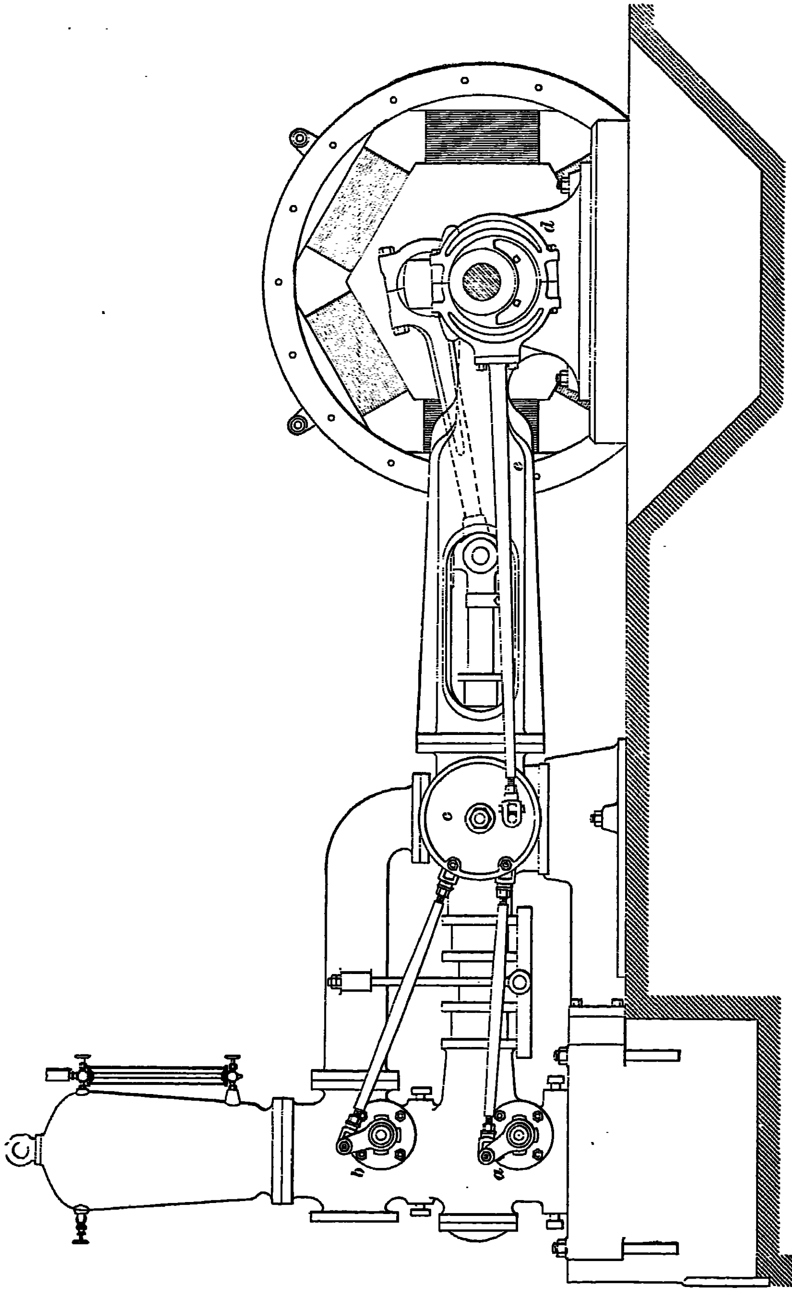
RIEDLER, ROTARY, AND VACUUM PUMPS

RIEDLER PUMPS

CONSTRUCTION OF RIEDLER PUMPS

72. Riedler pumps are the invention of Professor Riedler and are a type of pump designed for running at very high speeds. By study, experimenting, and careful noting of cause and effect, he discovered several very important phenomena. He found that there is much greater resistance to the flow of water through the valve passages in ordinary pumps than was before this thought to exist. He further found that the slip of ordinary valves is very large, and even when small has a tendency to cause severe hydraulic shocks throughout the pressure parts of the pump. He also was aware that the frictional resistance to the passage of a certain quantity of water through a large number of small openings is much greater than that existing when the same quantity of water passes through a single opening equal to the combined area of the smaller ones. With these facts in view, he designed a pump valve having the useful valve area as large as possible and containing as few separate passages as is consistent with good construction. He substituted one large valve for many small ones, thus decreasing the friction of the water in the valve passages. The reduction of the slip was accomplished by arranging a mechanical controlling device, whereby at the proper time and without restricting the water passage the valve was closed. The mechanical controlling device further assists in the reduction of friction in the valve passages, as it permits the valve lift to be high, thus increasing the effective area.

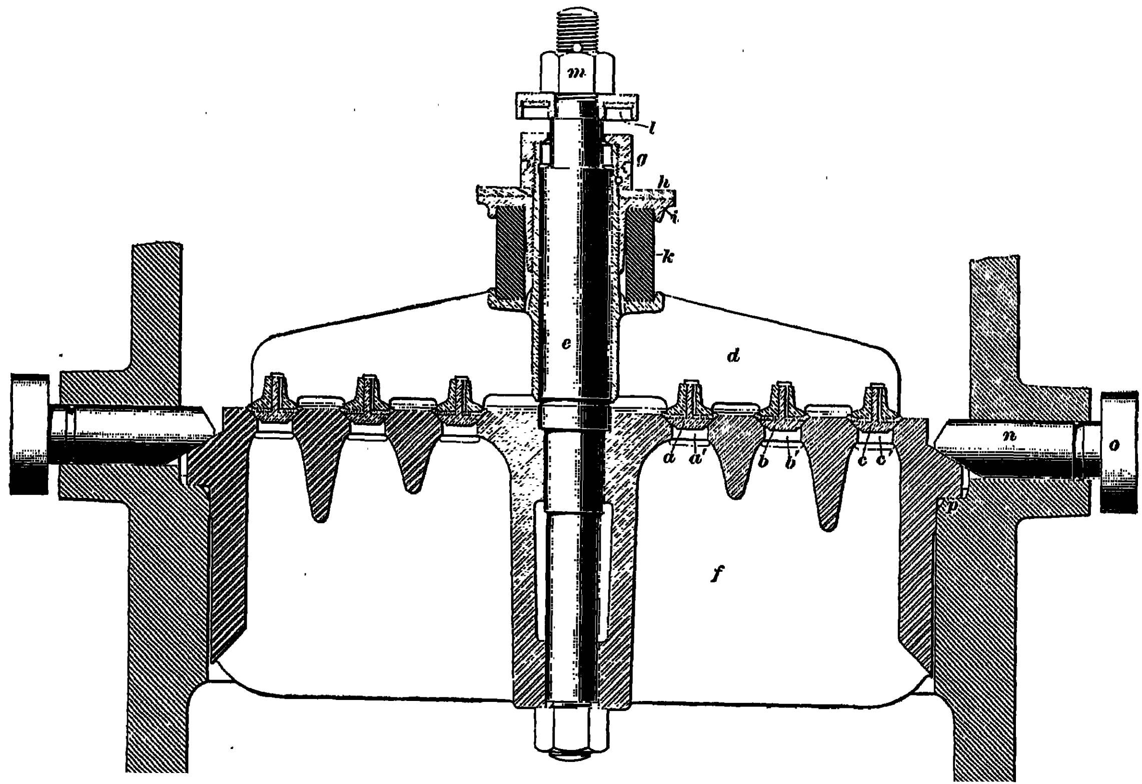
73. Fig. 41 shows an outside view of a direct-connected, electrically driven, differential Riedler pump having the plunger arrangement shown in Fig. 31 (*b*). The pump valves are



closed by cranks, the crank *a* operating the suction valve and the crank *b* the delivery valve. The two cranks are operated from a wristplate *c* similar to that of a Corliss engine and to which they are connected by the rods shown. The wristplate is rocked back and forth by the eccentric *d* on the crank-shaft, to which it is connected by the eccentric rod *e*. The plungers are driven by a crank, as shown.

74. Fig. 42 shows a detail of the improved Riedler suction and delivery valve. Both suction and delivery valves are alike in these pumps except as regards the flange for securing them to the pump chambers. The valve proper consists of three concentric bronze rings *a*, *b*, and *c*, each of which is cast in one piece and which are set into a spider *d* having eight arms. This spider is free to move up and down on the central valve post, or valve spindle, *e*. This valve rests on a heavy cast-steel valve seat *f* having three annular openings *a'*, *b'*, and *c'*. The valves proper are not rigidly connected to the spider, but each valve is free to form its seat with the valve seat and independent of the spider or each other. A leather ring between the valve proper and the spider serves to make an absolutely tight joint. A circular nut *g* is secured to the top of the hub of the valve spider *d* and holds in place a steel pressure plate *h*. This pressure plate rests on top of a spring cap *i*, below which a spring *k* of soft rubber is placed. This rubber allows of a certain amount of yield between the valves and their seat in case any foreign matter should get between them. Two steel fingers, not shown in the drawing, press upon the pressure plate and serve to close the valve just before the piston reaches the end of its stroke. A water cushion *l*, the object of which is to prevent the valve from striking its stop when opening, is secured to the top of the spindle *e* by the nut *m*. The nut *g* is closely fitted to the chamber in *l* and traps the water in front of it, thus making a hydraulic cushion. The valve seats are secured in the valve chambers by wedge-shaped plugs *n*, which are forced in by studs and nuts through the gland *o*, the effect being to force the valve seat *f* hard down on its bearing *p* in the pump chamber.

77



75. Fig. 43 is a perspective view of the Riedler valve and seat, showing the operating mechanism by means of which the valve is seated. All visible parts are lettered the same as in Fig. 42. The crank *q* is operated from the wristplate shown in Fig. 41; it is keyed to a shaft *r*, which passes through a stuffingbox *s* bolted to the valve chamber and carries a forked crank at its inner end. The jaws or fingers *t*, *t* of the forked crank press upon the pressure plate *h* to seat the valve at the

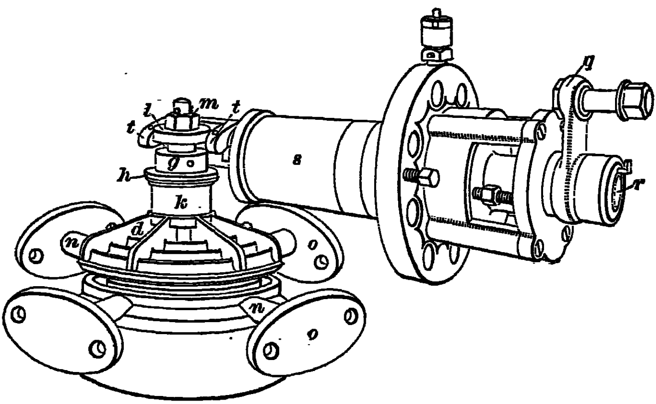


FIG. 43

proper time. The motion of the fingers is so timed in relation to the motion of the plungers that the fingers are clear of the pressure plate *h* when the plungers begin to deliver water, thus leaving the valve free to open.

76. The Riedler valve is by no means confined only to water pumps. It has been and is used successfully for high-pressure air and gas compressors. The Riedler pump may be driven by a steam engine, electric motor, turbine water-wheel, by belting, or in any other convenient manner.

RIEDLER EXPRESS PUMP

77. A type of Riedler pump that has been brought out for running at a very high speed is called the **Riedler express pump** and is shown in Fig. 44. Although the ordinary Riedler

pump can be run at speeds as high as 150 revolutions per minute and sometimes faster, conditions arise requiring a much higher speed, and to meet this condition this special design,

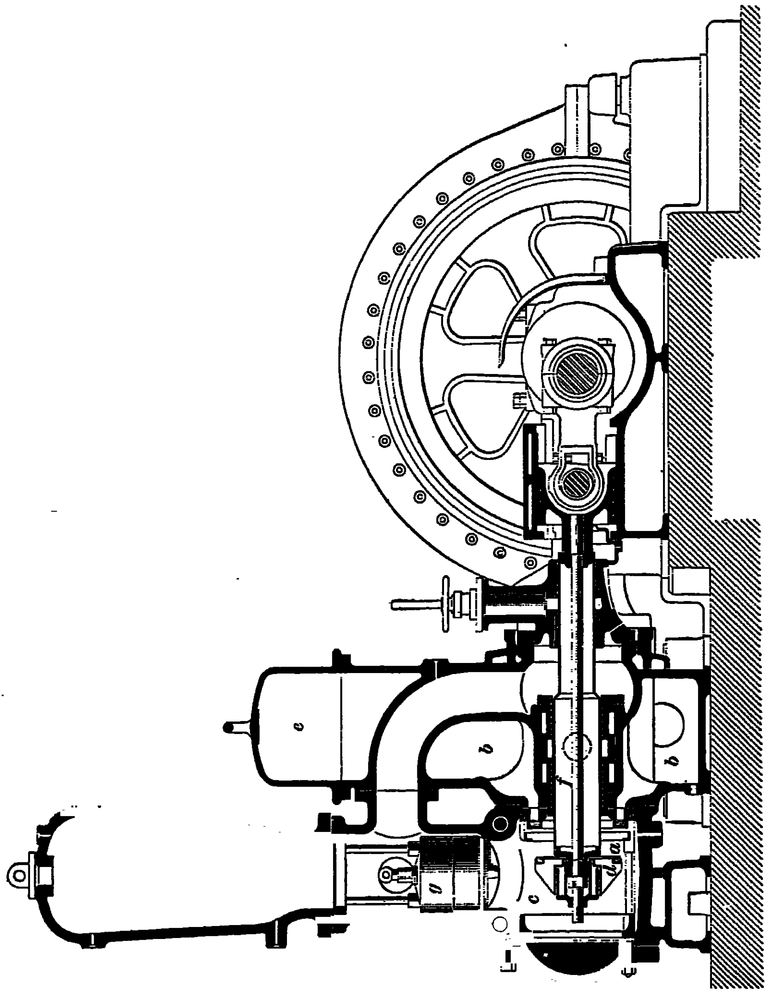


FIG. 44

which may be run at speeds as high as 300 revolutions per minute, has been developed. The main feature of this pump—in fact, the part that permits running at such high speeds, is

its suction valve. As will be seen by referring to the figure, the suction valve *a* is annular in form and is concentric with the plunger; it lifts in the direction opposite to that of the plunger when on its suction stroke, the water flowing from the suction chamber *b* into the valve chamber *c*. At the end of the suction stroke a buffer *d* mounted upon the end of the plunger drives the suction valve to its seat, making it certain that the valve is seated when the plunger starts on its delivery stroke and allowing practically no slip. A high suction air chamber *e*, containing a column of water, is placed above the suction valve, making it certain that the pump will fill as the plunger *f* makes its suction stroke. The delivery valve is shown at *g*. It will be noticed that this pump is of the differential type.

The chief point of advantage of the express pump is that it may be connected to high-speed motors. It is of small dimensions compared to the quantity of water it can handle, and thus consequently low in first cost.

ROTARY PUMPS

POSITIVE-DISPLACEMENT ROTARY PUMPS

78. Many attempts have been made to replace the reciprocating motion of the piston or plunger as used in the ordinary pump by a continuous rotary motion, thereby to do away with the necessity for suction valves and delivery valves. Some of the designs of rotary pumps that have been placed on the market have proved highly successful when used under favorable conditions.

In practice rotary pumps are divided into two general classes, which are *positive-displacement rotary* pumps and *centrifugal* pumps. In pumps of the first class the liquid to be pumped is displaced by positive means; in pumps of the second class the liquid is displaced by centrifugal force, and such pumps are not positive in their displacement. Thus, a centrifugal pump filled with water may be run with the water outlet entirely closed without creating an undue pressure; a positive-displacement

rotary pump cannot be run under such a condition, however, without breaking it.

79. One of the oldest forms of a positive-displacement rotary pump is the type known as the *gear pump*, an actual design of one being shown in Fig. 45. This form of pump has proved very successful for such work as forcing oil under pressure to the bearings of steam turbines, and for similar work where the liquid to be handled is free from grit and only a small quantity of liquid is to be pumped.

There are two meshing spur gears *a* and *b* on shafts having bearings in the housing *c*; the gear *a* is driven in the direction of the arrow marked on it and drives the gear *b*. The housing *c* closely fits one-half the circumference of each gear, and in conjunction with the two gears forms an inlet chamber at *d* and an outlet chamber at *e*. The inlet to the pump is at *f* and communicates through a port *g* with the inlet chamber *d*; the port *h* in the outlet chamber *e* communicates with the outlet *i*, from which a pipe leads the liquid to its destination.

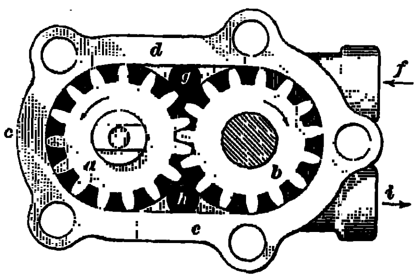


FIG. 45

Rotation of the gears in the direction of the arrows marked on them takes the liquid handled from the inlet chamber and carries it around the semicircular parts of the housing to the outlet chamber.

80. The **Roots rotary pump**, built by the P. H. & F. M. Roots Company, Connersville, Indiana, is a modification of the simple gear pump that was shown in Fig. 45. The essential parts of a Roots pump are shown in perspective in Fig. 46. It consists of two impellers *a* having straight sides and semicircular lobes *b*, the lobes being bolted to the straight-sided parts of the impellers, which parts are forged solid with the

shafts *c*. The two shafts *c* are geared together by spur gears on the outside of the pump casing *d*, which gears are not

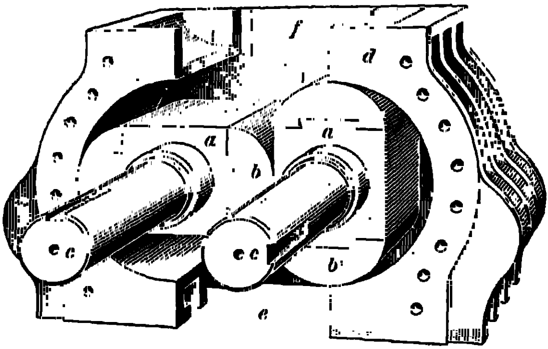


FIG. 46

shown; these gears force the impellers to turn at the same speed in opposite directions. The suction inlet of the pump

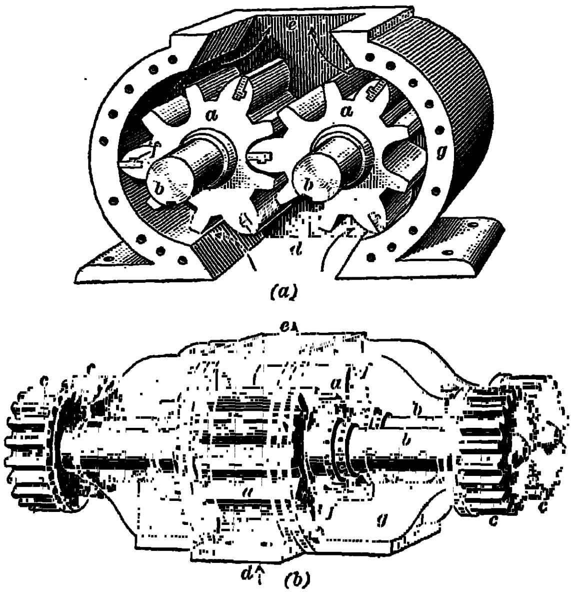


FIG. 47

is at *e*, and the discharge outlet at *f*. The Roots pump, unlike the simple gear pump, is well adapted to handle a large quan-

tity of water; pumps of this design have been built as large as 70,000 gallons per minute capacity.

81. The **Silsby rotary pump,** shown in Fig. 47 (*a*) in section and in (*b*) in perspective, which has been built for steam fire engines and other service by the American-La France Fire Engine Company, Elmira, New York, is a modification of the simple gear pump that was shown in Fig. 45. The pump has two three-lobed impellers *a* fastened to the shafts *b* which are forced to rotate at the same speed in opposite directions by the external spur gears *c*. The water inlet is at *d* and the outlet at *e*. Each lobe of each impeller is fitted with a removable packing strip *f*, which make water-tight joints between the impellers and also between the impellers and casing *g*.

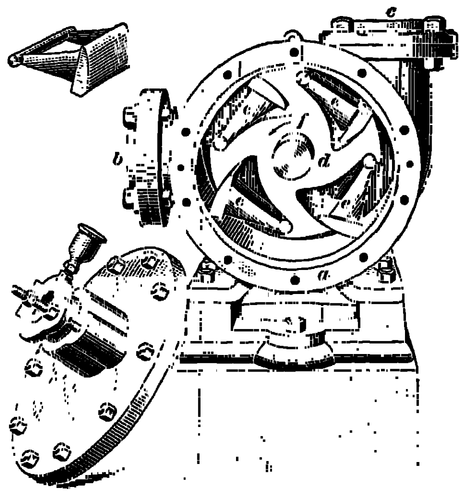


FIG. 48

82. The **Blackmer rotary pump,** built by the Blackmer Rotary Pump, Power, and Manufacturing Company, Petoskey, Michigan, which is shown partly disassembled in Fig. 48, consists of a circular casing *a* having the water inlet at *b* and the water outlet at *c*, and a rotating piston *d* placed eccentric in reference to the casing. The piston has four recesses, as shown, in which are mounted the movable buckets *e*; one such bucket is shown separately in perspective. As the piston *d* rotates in the direction of the arrow *f*, the four buckets are kept against the upper part of the circular casing by centrifugal force, and against the lower part by centrifugal force and gravity, and thus each bucket in passing the inlet traps a certain quantity of water in the lower part of the

casing which is carried around to, and discharged from, the outlet.

83. In the **Quimby screw pump,** shown in Fig. 49 and built by the Quimby Company, New York City, New York, there are two parallel shafts *a* connected by the gears *b*. Each shaft has a right-handed and a left-handed screw, and

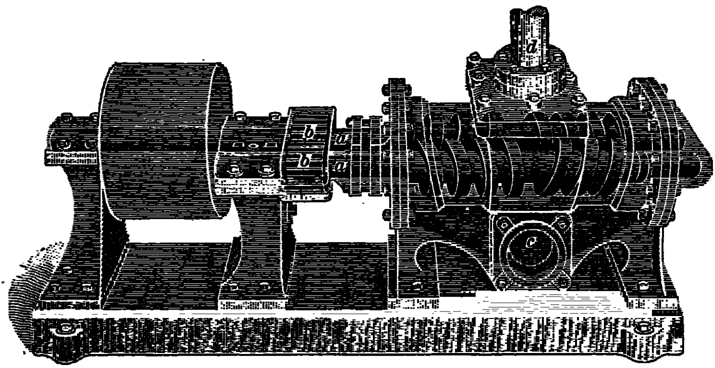


FIG. 49

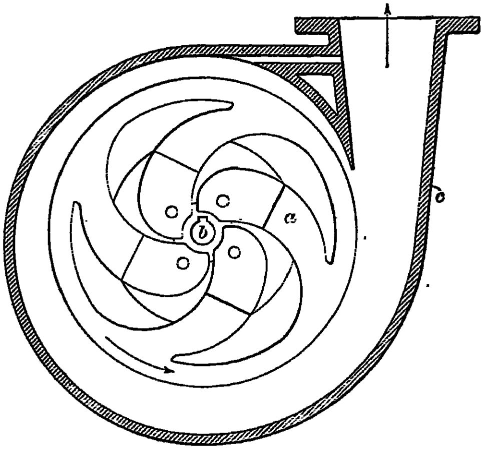
each screw meshes with the other, so that water coming through the suction pipe at *c* flows into the casing and to the outer ends of the screws, being drawn toward the center as the screws revolve and finally discharging through the discharge opening at *d*. The screws themselves fit each other and the pump casing closely. This type of pump may be run either by belt or by electric motor, or it may be driven by a direct-connected engine.

CENTRIFUGAL PUMPS

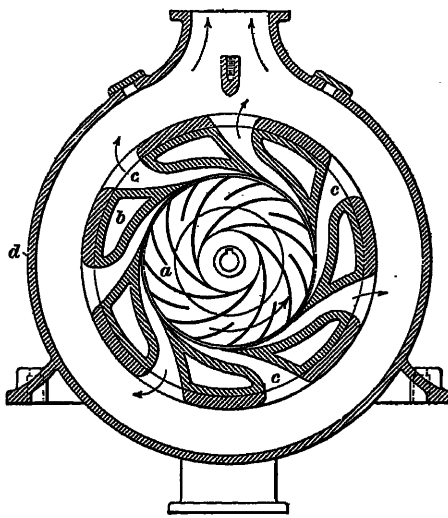
84. Definitions and Classifications.—Any pump the action of which depends on the pressure produced, through centrifugal force, by a quantity of water rotated very rapidly by means of curved vanes on a rotating shaft, is called a *centrifugal pump*.

There are two general classes of centrifugal pumps, which are known respectively as *volute pumps* and *turbine pumps*. The principle of construction of the volute pump is shown in Fig. 50 (*a*). The pump consists essentially of two parts, which

are the *impeller* *a* fastened to the shaft *b*, and the casing *c* having a spiral form either conforming to or approximating



(a)



(b)

FIG. 50

the curve known as the *volute*, whence the pump derives its name. The pump casing may have a form different from that

of a volute; it may even be circular, but experience has shown that with a casing differing much from the volute form the pump will be very inefficient. The impeller is rotated from any convenient source of power by belt, chain, or rope drive, or is direct-connected to an engine or electric motor; it has a number of blades, or vanes, curving backwardly in relation to its rotation, and is also called by various other names, such as *rotor, runner disk, runner, flyer,* or *fan.* The water inlet is at the center of the impeller, and water entering the revolving impeller is carried around with it and at the same time travels outwards toward the periphery of the impeller, whence it is discharged at a high velocity into the volute of the casing and passes onward to the discharge outlet. The velocity of the water leaving the impeller is gradually reduced in the volute part of the casing, and the kinetic energy of the moving body of water is thus transformed into pressure.

The turbine pump, shown at (*b*), consists essentially of three parts, which are the impeller *a*, the diffusion ring *b* having a number of channels *c* curving forwards in relation to the direction of rotation of the impeller, and the casing *d*, which may be circular, as shown, or may be composed of two volutes. The diffusion-ring channels gradually reduce the velocity of the water leaving the impeller and thereby change the kinetic energy of the moving body of water to pressure.

85. Centrifugal pumps have no valves or restricted passages to hinder the flow of a liquid through them, and are therefore serviceable not only for pumping hot or cold liquids but can be used also for pumping water containing large quantities of mud, sand, gravel, or anything else that can be carried through the pump and pipes by a current of water. Water containing much foreign matter in suspension cannot be handled successfully by reciprocating pumps or rotary pumps of the positive displacement type, as the solid matter would soon destroy the working parts or prevent closing of the valves. Centrifugal pumps of the volute type when used for sand dredging and similar work where the liquid to be pumped contains sharp quartz or any other material whose abrasive action would soon

cut out a cast-iron pump, often have the casing fitted with a steel lining, which may be a manganese-steel casting, this material being so hard that it cannot be machined except by grinding. The impeller in such a case is made of the same material as the lining of the casing. In a lined centrifugal pump the lining may be replaced by a new one when worn out, this repair being cheaper than replacement of the whole pump.

For handling corrosive liquids centrifugal pumps of both the turbine and the volute types are often made entirely of special corrosion-resisting compositions, instead of being lined.

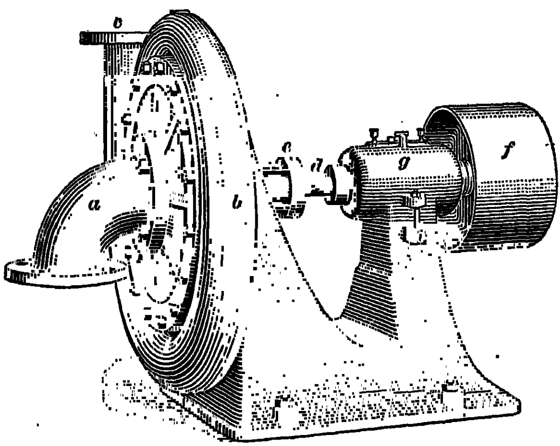


Fig. 51

86. Centrifugal pumps of both the turbine and the volute types are classified as *single-suction* and *double-suction pumps.* In a single-suction pump the water enters at one side of the impeller, and consequently exerts a side thrust on the impeller, which must be resisted by suitable means; in a double-suction pump the water enters both sides of the impeller, and consequently there is no side thrust. For pumping against fairly low pressures centrifugal pumps are usually made **single-stage,** which means that the pump has only one impeller; for pumping against high pressures, centrifugal pumps are made **multistage,** which means that the pump has a number of impellers which operate in succession on the liquid

pumped. In a multistage centrifugal pump the discharge of the first impeller passes under pressure to the inlet around the center of the second impeller and leaves its periphery at an increased pressure, passing thence to the center of the third impeller, and so on. The number of stages used depends on the final pressure to be pumped against; sometimes ten impellers are employed. Single-stage and multistage centrifugal pumps are built in both the single-suction and double-suction types.

Centrifugal pumps are spoken of as *horizontal pumps* when the shaft to which the impeller is fastened is horizontal, and

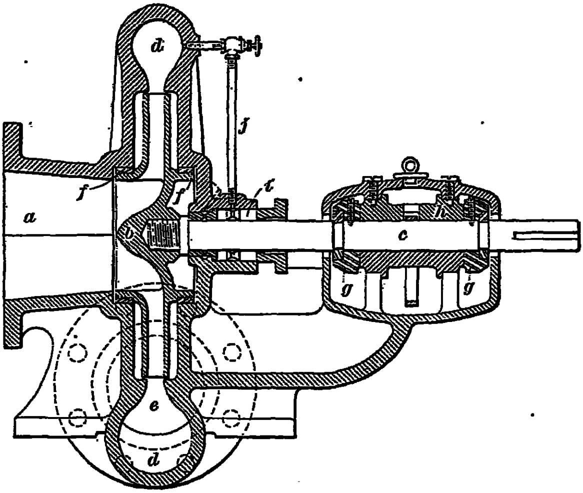


FIG. 52

as *vertical pumps* when the shaft just referred to is vertical. When a centrifugal pump is placed directly into the liquid it pumps, it is called a *submerged pump*.

87. Construction of Centrifugal Pumps.—An external perspective view of a single-suction single-stage volute centrifugal pump of the horizontal type is shown in Fig. 51; the pump shown is arranged to be driven by a belt from some convenient source of power. The water inlet for the pump is shown at *a*, which leads to the center of the impeller. The volute part of the casing *b*, into which the impeller discharges

the water, has a circular cross-section increasing in size toward the water outlet *c*. The impeller shaft *d* passes through a stuffingbox at *e*, which is usually packed with a soft fibrous packing, and carries the belt pulley *f*. Alinement of the impeller in reference to the casing is preserved by constructing the bearing *g* so that it acts as a thrust bearing.

88. A sectional view of a single-suction single-stage centrifugal volute pump made by the Jeanesville Iron Works Company, Hazleton, Pennsylvania, is shown in Fig. 52. The water

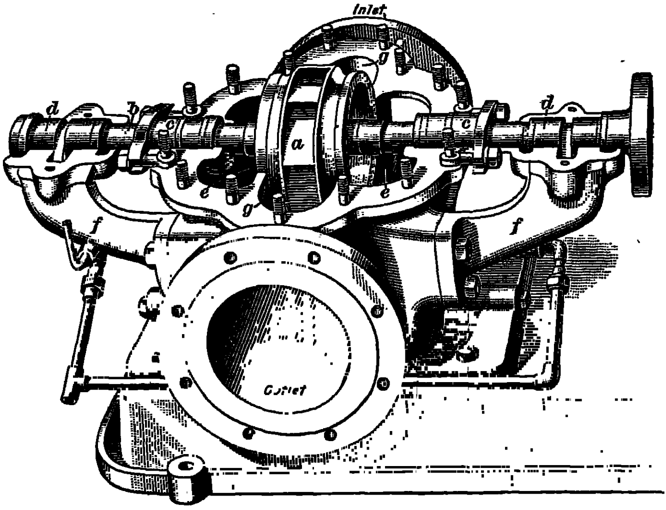


FIG. 53

inlet *a* delivers the water to the central part of the impeller *b*, which is fastened to the shaft *c*; the water is delivered into the volute part *d* of the casing and discharged from the outlet *e*. Leakage of water from the discharge side to the inlet side of the pump is prevented by the impeller bushing rings *f*, which are a close fit in machined recesses of the casing. To preserve the axial alinement of the impeller, the shaft *c* is fitted with the two large collars *g*, one being placed on each side of the bearing *h*. In this case the stuffingbox on the casing through which the impeller shaft passes is fitted with a metallic water-seal ring *i*, placed between rings of fibrous packing; through the

pipe *j* water under pressure is admitted from the volute part of the casing to the water-seal ring, the water surrounding the impeller shaft and thereby preventing any entrance of air to the inlet side of the pump.

89. A perspective view of a double-suction single-stage volute pump made by the De Laval Steam Turbine Company, Trenton, New Jersey, is shown in Fig. 53 with the top half of the casing removed; a cross-sectional view of the same pump

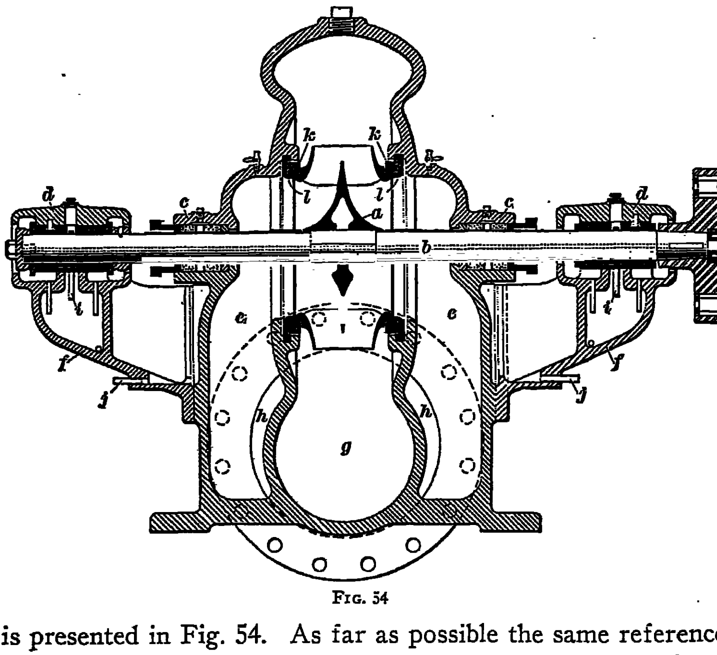


Fig. 54

is presented in Fig. 54. As far as possible the same reference letters have been given to the same parts in both illustrations, and both should be referred to in reading the description. The impeller *a* is a bronze casting fastened to the steel shaft *b* which passes through the two stuffingboxes *c* and is supported by the two ring-oiled bearings *d*. These bearings are made independent of the stuffingboxes *c*, to prevent the impeller *a* and the shaft *b* from getting out of alinement through wear, and also to prevent the lubricant from entering the pump chamber *e*. Within the latter the shaft *b* is surrounded by renewable bronze sleeves.

These sleeves are screwed into the impeller, so as to prevent the liquid that is being pumped from coming in contact with the shaft. To prevent entrance of air around the shaft and leakage of a cold liquid, cupped-leather packing rings, which are set out or held tightly to the shaft by the pressure of the liquid, are placed in the stuffingboxes. When pumping a hot liquid, however, graphited flax packing is substituted for the leather packing rings.

The impeller is of the enclosed type. It takes in water from the suction, or inlet, chambers *e* at the center, on both sides of its hub, thus preventing end thrust; the water enters these two chambers from the single water inlet *h*. The water is discharged through the passages between the curved vanes, or blades, to the periphery and into the outlet, or discharge, chamber *g* terminating in the delivery pipe.

The brackets *f* that carry the bearings *d* form oil reservoirs at their outer ends, which reservoirs contain oil into which dip the oil rings *i* that by their rotation with the shaft *b* carry oil up to the bearings. The inner ends of the brackets *f* form drip boxes that catch any leakage of water from the stuffingboxes; drain pipes *j*, Fig. 54, convey any drip water to some convenient source of disposal.

Leakage from the discharge side of De Laval centrifugal pumps to the inlet side, past the sides of the impeller, is prevented by grooved packing rings, one pair of rings *k*, Fig. 54, being carried by the impeller *a*, while a pair of stationary rings *l* are carried by the casing. The circular grooves of the rings *k* and *l* interlock and are a fairly close fit; the grooves form a very tortuous path for water to leak through, and owing to this tortuous path the packing rings are said to constitute a *labyrinth packing*.

90. A multistage single-suction volute centrifugal De Laval pump is shown in section in Fig. 55; the impellers, etc. removed from the pump are shown in perspective in Fig. 56. The same parts have been given the same reference letters in both illustrations, and both should be referred to in reading the description. The pump shown is of the three-stage type, there

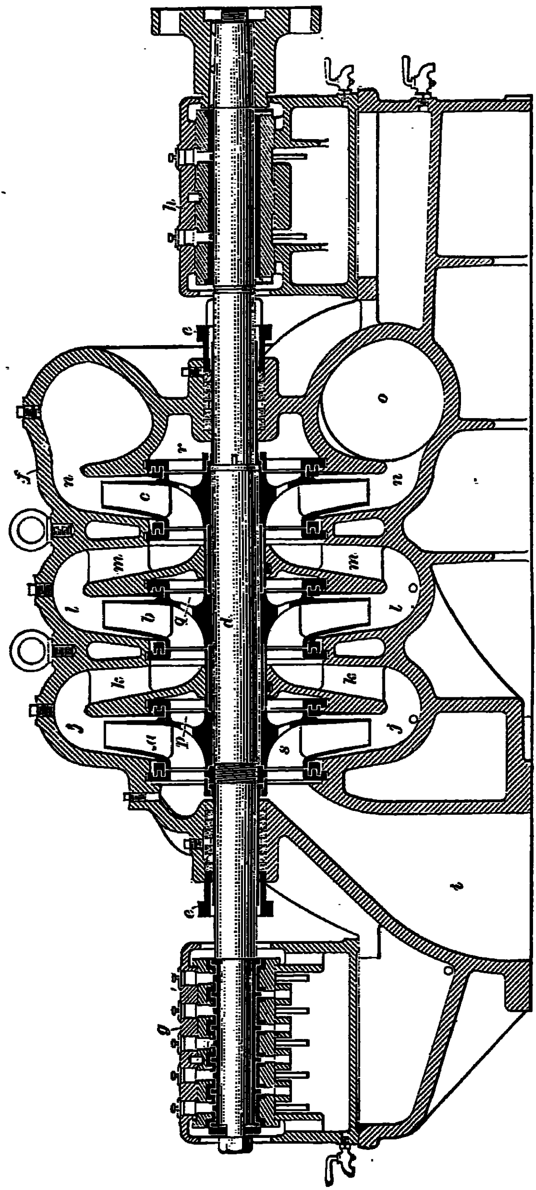


FIG. 55

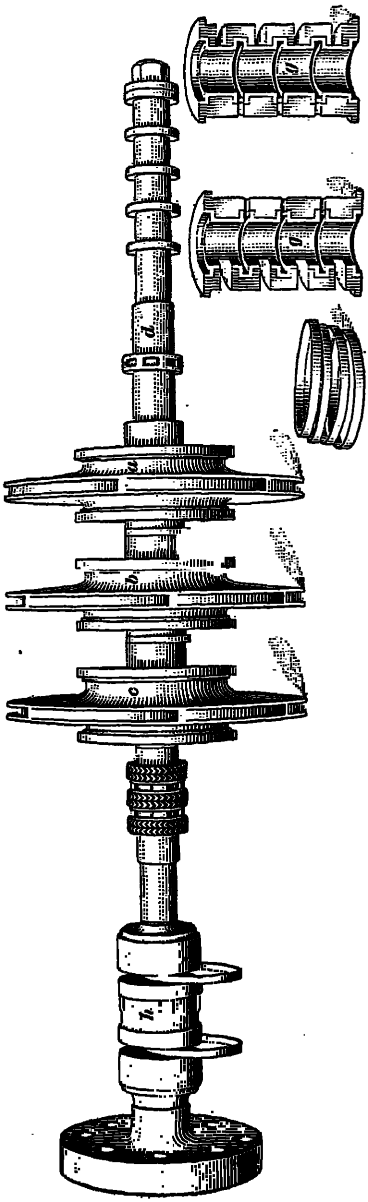


FIG. 56

being three impellers *a, b, c,* each of which receives water on one side. The three impellers are fastened to the rotor shaft *d,* which passes through the two stuffingboxes *e* of the casing *f* and is supported in the two bearings *g, h*; the two halves of the bearing brasses *g* are shown separate in Fig. 56. The bearing *g* has a number of grooves cut in it, into which fit corresponding collars on the impeller shaft, and endwise movement of the impellers is thereby prevented. The water inlet of the pump is at *i*; the water enters at the center of the impeller *a* and is discharged into the chamber *j* of the casing, whence it passes under pressure through the channels *k* to the center of the impeller *b*. This impeller discharges into the chamber *l,* whence the water under an increased pressure passes to the third impeller *c* through the channels *m* and is discharged into the last chamber *n* and passes thence to the discharge opening *o* under its final pressure.

End thrust due to the water entering on one side of each impeller is neutralized by means of the *balancing chambers p, q,* and *r,* Fig. 55. These chambers are bounded by partitions of the casing *f* on one side and the web of the impellers on the other side; the web of each impeller is pierced by a number of large holes, as *s,* through which the pressure on the inlet side of each impeller is transmitted to the water in the balancing chambers. Since the same pressure is thus exerted on the same area of both sides of each impeller, the end thrust is eliminated. A pair of labyrinth packing rings on each side of each impeller reduce to a minimum leakage of water from the discharge side to the inlet side of each impeller.

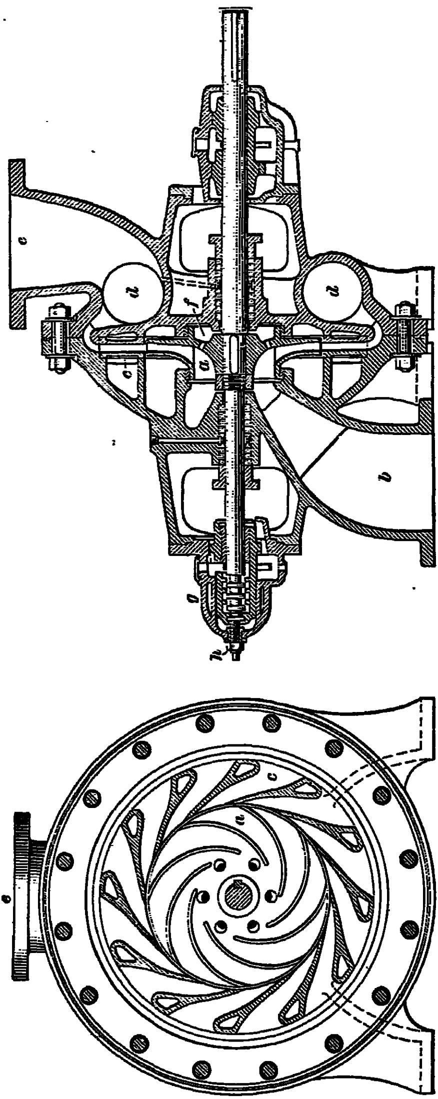


Fig. 57

91. A single-stage single-suction turbine pump made by the Alberger Pump Company, New York City, is shown in Fig. 57. The single impeller *a* receives the water

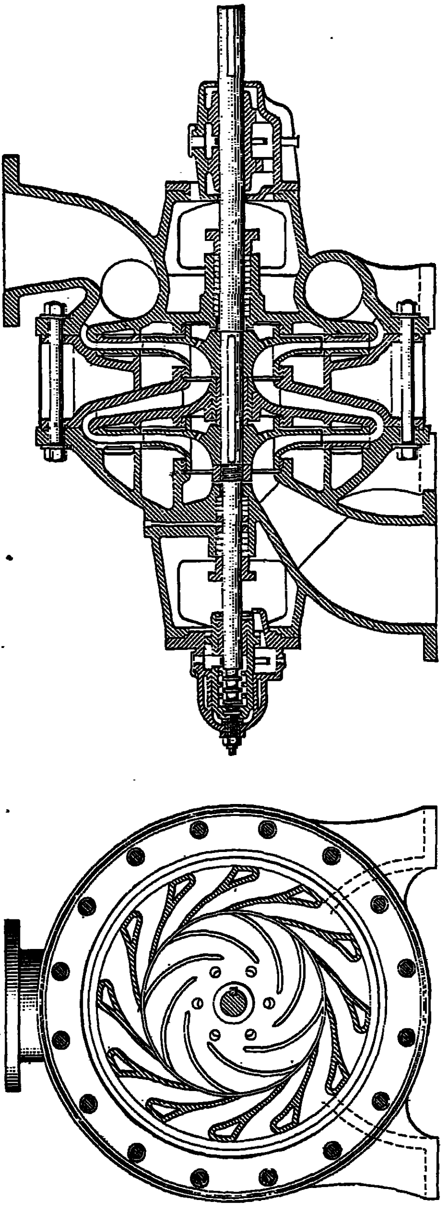


Fig. 58

from the inlet *b* and discharges it into the diffusion ring *c* having a number of diffusion channels formed by the vanes of the partitions shown. From the diffusion ring the water under pressure passes into a circular concentric chamber *d* and thence to the discharge outlet *e*. End thrust due to the water entering on one side of the impeller is eliminated by the balancing chamber *f*, and endwise alinement of the impeller is preserved by collars on the shaft engaging grooves in bearing *g*, and also by the thrust-adjusting screw *h*.

92. The multi-stage single-suction Alberger turbine pumps, one of which is shown in Fig. 58, differ from the single-stage pumps in having several impellers in series, as an examination of the illustration will reveal.

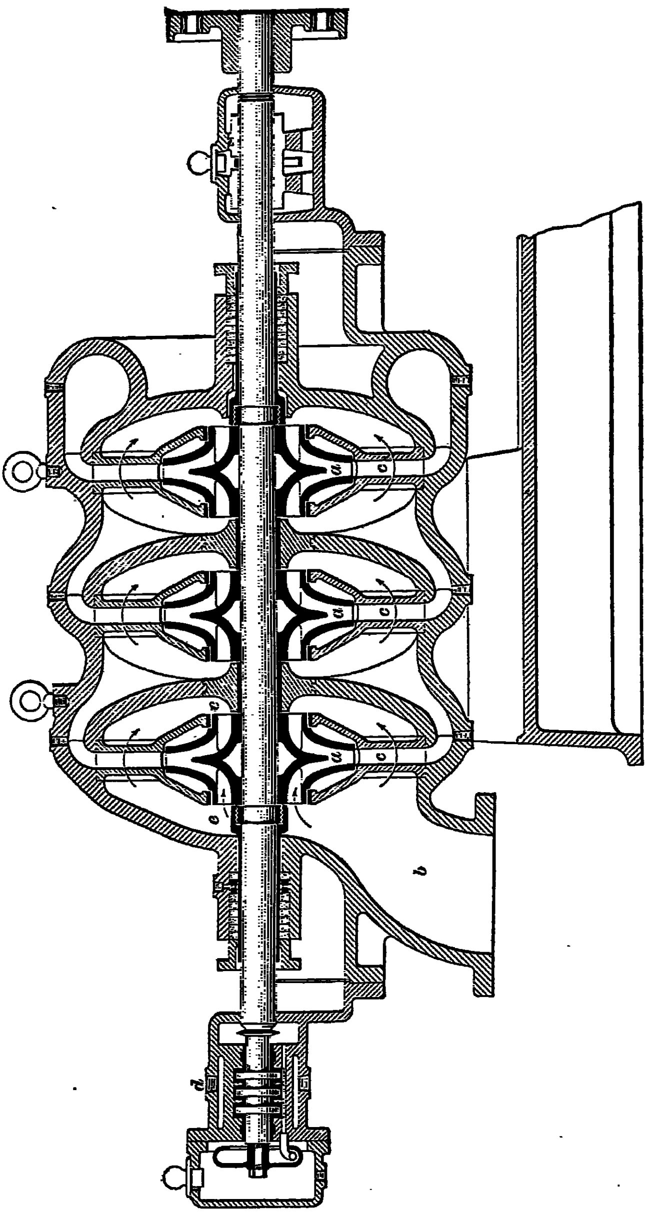


FIG. 59

While the turbine pumps shown in Figs. 57 and 58 are of the single-suction type, several manufacturers of turbine pumps employ a double-suction design.

A three-stage double-suction turbine pump, made by the Jeanesville Iron Works Company, Hazleton, Pennsylvania, is shown in cross-section in Fig. 59. There are three double-inlet impellers *a*; the water to be pumped enters at *b* and is free to pass to the right-hand side of each impeller through a number of large holes in each diffusion ring *c*, as indicated by the arrows. These holes are located between the diffusion-ring vanes that form the channels through which passes the water discharged by the impellers. Endwise alinement of the impellers is preserved by collars on the impeller shaft which engage corresponding grooves in the bearing *d*. A water-tight joint between each impeller and diffusion ring is made by renewable bushing rings *e*.

93. When a very large quantity of water is to be pumped to a moderate height by a centrifugal pump, this may be done by a single very large pump driven by a steam engine or other slow-speed prime mover, or it may be done by using a number of small pumps all connected to the same suction pipe and all discharging into the same discharge pipe; pumps thus connected are said to be *in parallel.* When small pumps are chosen, they are usually driven by a steam turbine or directly by an electric motor, which two types of prime movers are essentially high-speed machines.

Sometimes several separate pumps in parallel are arranged to be driven simultaneously by the same source of power; in other cases several impellers are combined into a single casing, all impellers being in parallel. Pumps constructed in the manner last named are called *multi-impeller centrifugal pumps,* and are usually of the single-stage type. The chief advantage of using small pumps in parallel, rather than a single large pump and slow-speed source of power, is the low cost of the installation.

94. An example of two centrifugal volute pumps of the double-suction type connected in parallel is presented in Fig. 60,

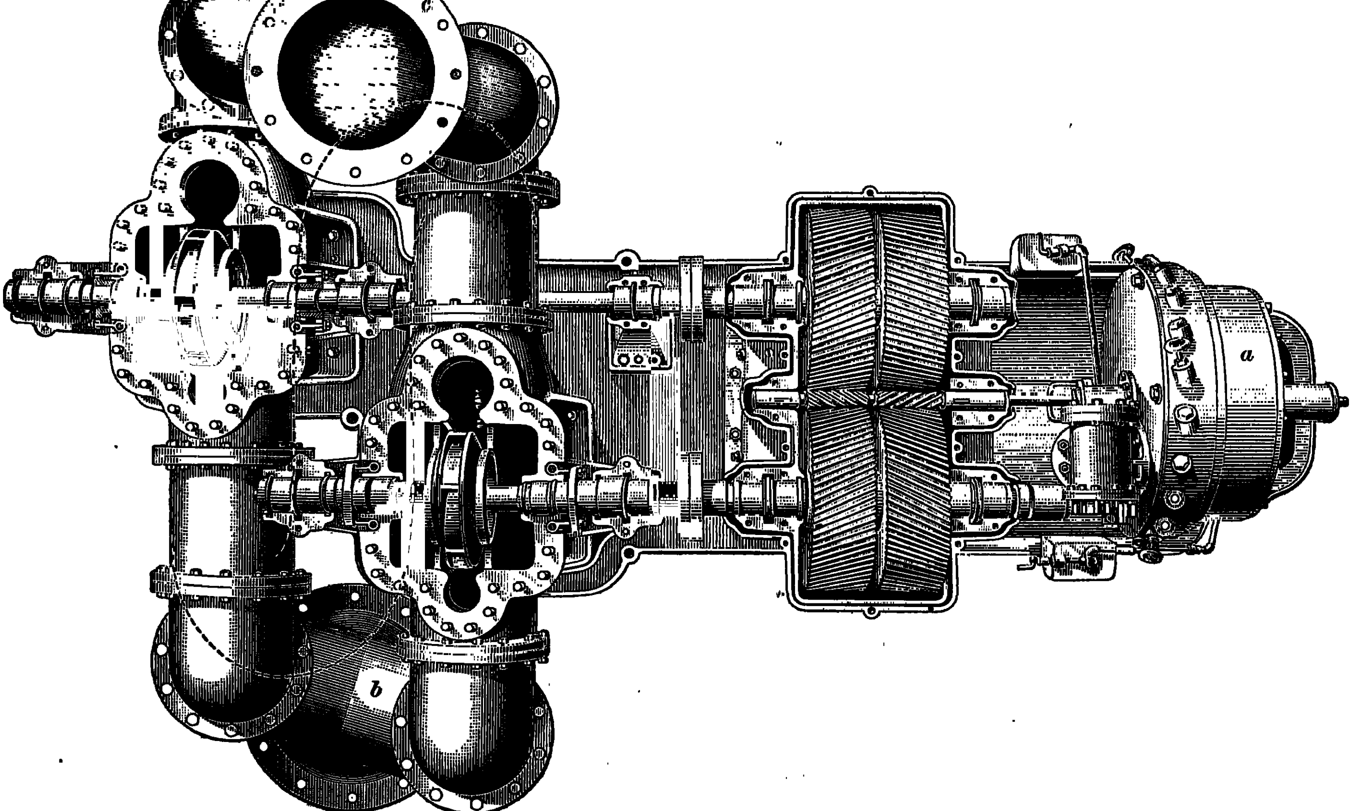


FIG. 60

where a De Laval single-stage steam turbine *a*, through two large reduction gears, drives the impellers of the two pumps shown with the upper half of their casing removed. The two water inlets are connected by elbows to a **Y** fitting *b* to which the common inlet pipe is bolted; the water outlets of the two pumps are connected to a **Y** fitting *c* to which the common discharge pipe is bolted.

95. An example of a multi-impeller single-stage double-suction turbine centrifugal pump with three impellers, built by the Alberger Pump Company, New York City, and driven by a single-stage Curtis steam turbine, is shown in perspective in

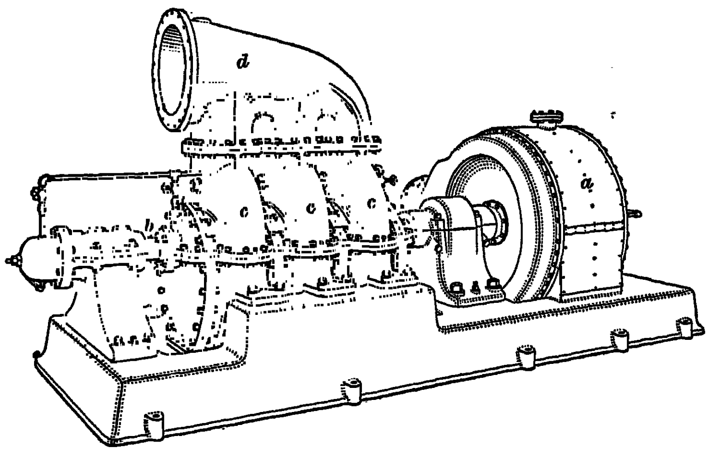


Fig. 61

Fig. 61. In this illustration the steam turbine *a* is direct-connected to the impeller shaft *b*; the three water outlets of the pump casing *c* are connected to the discharge manifold *d*. There are three water inlets, one for each impeller, which are connected to a single inlet manifold; owing to the view taken, neither the water-inlet manifold nor the water inlets can be seen in the illustration.

96. There are five different forms of impellers used in centrifugal pumps; they are *open-vane impellers, single side plate impellers, double side plate* or *enclosed-vane impellers, central-disk impellers,* and *hollow-arm impellers.*

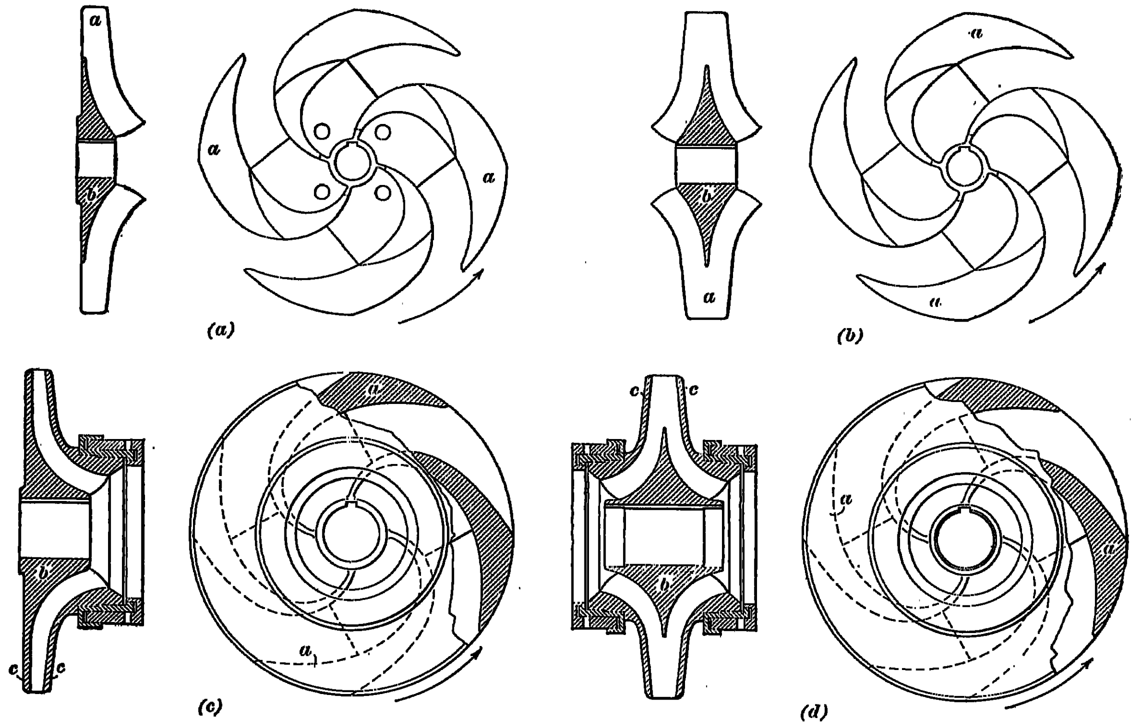
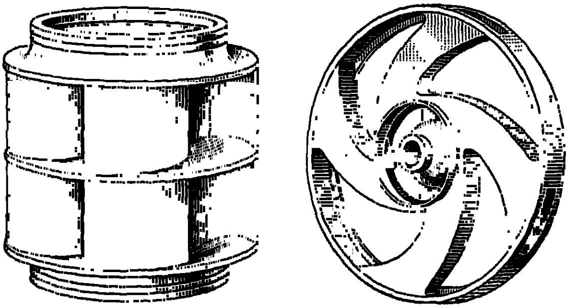


FIG. 62

An open-vane impeller for a single-suction centrifugal pump is shown in Fig. 62 (*a*), and for a double-suction pump at (*b*). This form of impeller has a number of vanes *a*, ranging from two upwards, which are cast in one with the hub *b*; the vanes are not enclosed in any way, but are made to conform closely to the shape of the part of the pump casing they run in. A single side plate impeller, as implied by the name, has the vanes cast in one with a plate on the side opposite the water inlet of the pump, and is used only with single-suction pumps of small size. A double side plate, or fully enclosed, impeller for a single-suction pump is shown in Fig. 62 (*c*), and for a double-



(*a*) (*b*)

FIG. 63

suction pump in (*d*); the vanes *a* of the impellers are cast in one with the hub *b* and side plates *c*.

A central-disk enclosed impeller is shown in Fig. 63 (*a*); this form of impeller is used for double-suction pumps intended to handle rather heavy liquids, the central disk serving greatly to strengthen the impeller. Central-disk impellers with open vanes, made for single-suction as well as for double-suction pumps, are also in use. A hollow-arm impeller is shown in perspective in (*b*); this form of an enclosed impeller is used for single-suction pumps by some pump makers in preference to a double-side plate impeller on account of its lighter weight.

97. Centrifugal pumps that are not submerged in the liquid to be pumped, or that are situated where the water can-

not flow into the inlet by gravity, must be *primed,* that is, filled with the liquid to be pumped, before they can start pumping.

Various means of priming centrifugal pumps are employed in practice. Thus, the inlet pipe may have a check-valve, known as a *foot-valve,* below the water level and opening toward the pump; in this case the pump and inlet pipe may be filled with water either from a water main or by pouring the water in with a bucket through a suitable valve on top of the pump casing. Or, a steam ejector or a small hand pump taking its water from the same supply as the centrifugal pump may be used to prime the pump. When no foot-valve is fitted, a check-valve or a shut-off valve may be placed in the discharge pipe, the check-valve opening toward the discharge outlet. In these cases the pump may be primed by using a steam ejector or an air pump to exhaust air from the pump casing, thus permitting atmospheric pressure to force water into the pump casing; if a shut-off valve in the discharge pipe is fitted, this must be closed before priming and opened again after the pump has been started. With a centrifugal pump a dangerous over-pressure is not created by starting it with the discharge shut off; the impeller simply churns the water until the discharge valve is opened.

98. The probable discharge of a centrifugal pump, in United States gallons per minute, can be estimated for a single-suction pump by multiplying the area of the discharge pipe in square inches by 34, and for a double-suction pump by multiplying that area by 37. By speeding up the pump, just as is the case with reciprocating and rotary pumps, a higher discharge rate is obtained, and conversely, by slowing the pump down the discharge rate is lowered; therefore, the capacity calculated as indicated here must be considered as an approximation.

VACUUM PUMPS

DEFINITIONS

99. Any pump designed to pump air, vapor, or gases from a closed vessel, and to maintain therein a pressure less than that of the atmosphere, that is, a partial vacuum, is called a *vacuum pump*. This class of pumps is divided into two general subclasses, which are *wet-vacuum pumps* and *dry-vacuum pumps*.

Wet-vacuum pumps are pumps that are required to remove continuously a considerable quantity of water from the space in which the vacuum is being maintained, as, for instance, from a jet condenser or surface condenser applied to a steam engine, steam turbine, etc.

Dry-vacuum pumps are pumps that do not handle any liquid excepting moisture that may be carried in suspension by the air or other gases in the vessel in which a vacuum is to be maintained; such pumps are used to some extent in connection with surface condensers for steam turbine work, where a very perfect vacuum must be maintained for the sake of economy in the use of steam. Dry-vacuum pumps have their greatest field in various lines of industry where liquids have to be evaporated under a pressure less than that of the atmosphere.

Vacuum pumps, in accordance with their construction, may be classified as reciprocating vacuum pumps, turbo vacuum pumps, and ejector vacuum pumps.

RECIPROCATING VACUUM PUMPS

100. A reciprocating vacuum pump is usually of the piston type, and where only a rather imperfect vacuum is required, may be an ordinary single or duplex direct-acting steam pump. When a very high vacuum is required to be maintained, however, a pump of the crank-and-flywheel pattern, or power-pump pattern, must be employed, because the definite length of stroke of the piston of such patterns of pumps permits the pump end to be designed with a minimum clearance volume, which is a consideration essential to the

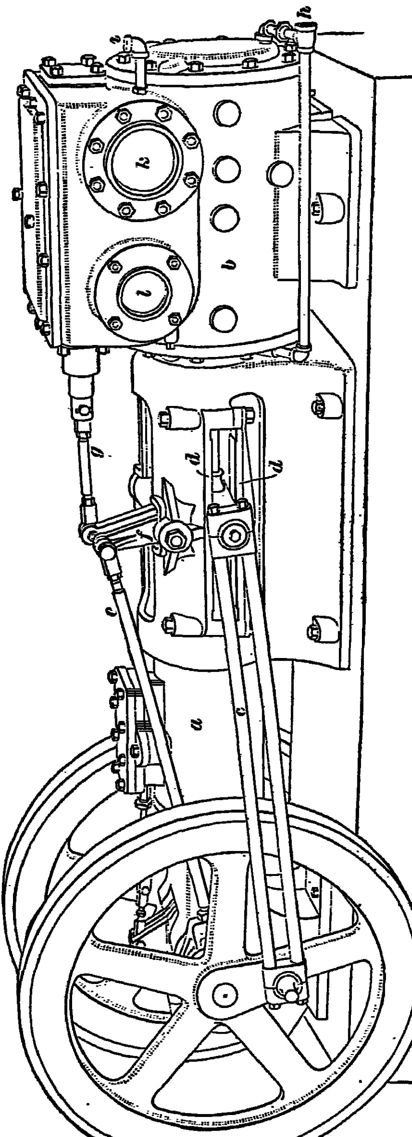


Fig. 64

production of a high vacuum. With dry-vacuum reciprocating pumps the moisture in the vapor to be pumped is often at so high a temperature that it expands into vapor when the pressure is sufficiently reduced, thereby reducing the vacuum that could otherwise be obtained, or increasing the work for the pump. To overcome this objectionable feature, a small amount of cold water is often injected into the pump suction to increase the vacuum by condensing part of the vapor, by cooling the vapor and thus reducing its pressure, and by filling the clearance volume with water so that on each suction stroke of the piston there is but little vapor left to expand and prevent prompt beginning of the suction.

101. An example of a crank-and-flywheel type of reciprocating dry-vacuum pump, built by the Deane

Steam Pump Company, Holyoke, Massachusetts, is shown in Fig. 64. This particular pump is characterized by the absence of the usual suction and delivery valves in the pump end, a mechanically operated slide valve controlling the intake and delivery of air. The steam cylinder *a* drives the crank-shaft through a connecting-rod, as in an ordinary steam engine, and is in line with the pump cylinder *b*. The pump piston is driven from the crank-shaft by two connecting-rods, as *c*, one on each side of the pump, which attach to a crosshead on the end of the pump piston rod, the crosshead being guided in a straight line by guides *d*. An eccentric on the crank-shaft is attached by means of an eccentric rod *e* to a double rocker-arm *f*, which in turn by means of the valve rod *g* drives the slide valve inside the valve chest of the pump cylinder *b*. The cylinder heads and the pump cylinder barrel are water-jacketed, cold water being circulated through the jackets to keep the pump cool; the cooling water enters at *h* and leaves at *i*. The suction pipe is attached to the flange *k*, and the discharge pipe to the flange *l*

102. Two cross-sectional views of the pump cylinder of a dry-vacuum pump of the type shown in Fig. 64 are presented in Fig. 65. There are two ports *a*, *b* connecting the valve seat with the ends of the pump cylinder; each of these two ports serves alternately as an inlet and a discharge port. The slide valve *c* contains two discharge ports *d*, *e*; the upper ends of the valve ports *d*, *e* are closed by a flat steel spring *f* serving as a valve that opens up against the stop *g*. The springs *f* open into the valve chest whenever the pressure in the pump cylinder, on the discharge stroke of the pump piston, exceeds the pressure of the atmosphere, and when at the same time the port *d* registers with the port *a*, or the port *e* registers with the port *b*. The air discharged through the valve *c* passes from the valve chest to the discharge pipe. When the pump piston is near the end of its stroke, the pressure on the one side is nearly that of the atmosphere, and on the other side, that of the vacuum created by the pump. At this time the valve *c* is brought to the position shown, where the so-called *flash port h* connects the ports *a* and *b*; the pressures on both sides of the

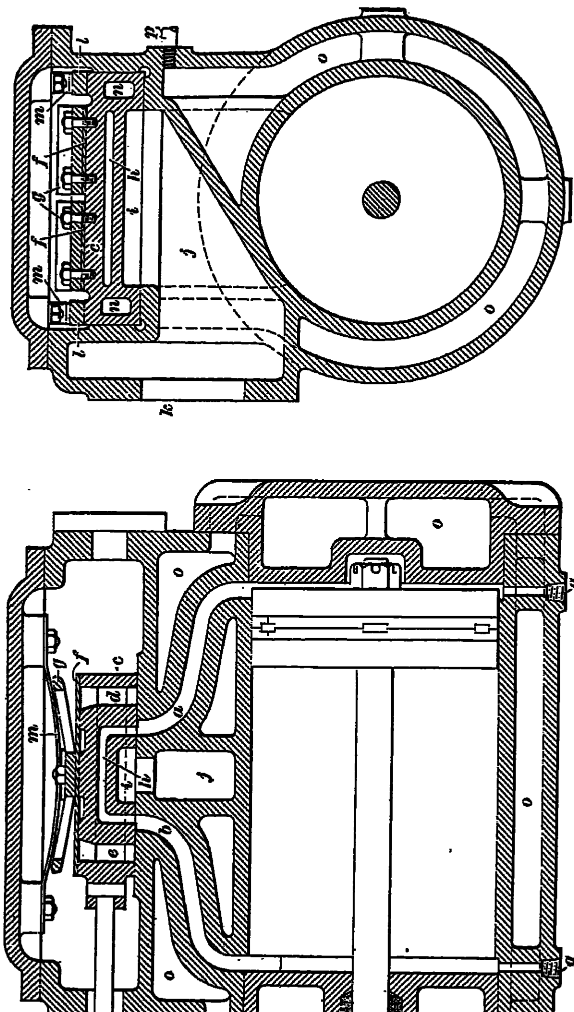


Fig. 65

pump piston promptly equalize, the resulting pressure being but slightly above that of the vacuum previously existing on the suction side. The lowering of the pressure in the clearance spaces before the piston begins a suction stroke is a decided advantage in that it means a prompt beginning of the suction. As soon as the pump piston begins its suction stroke, the valve *c* is shifted so that either the port *a* or the port *b* is connected by the valve cavity *i* with the suction inlet cavity *j*, to which the suction pipe is attached at *k*. The slide valve *c* is held to its seat by flat strips *l* and flat springs *m*. Two ports *n* cored in the valve *c* connect the valve discharge ports *d* and *e*, so that no matter which end of the cylinder is discharging, both ends of the flat steel spring *f*, which serve as discharge valves, can open. The jacket spaces are shown at *o*, the water outlet being at *p*. Cylinder drain cocks are attached at *q*.

103. Reciprocating vacuum pumps when applied to condensers for steam engines, steam turbines, etc. are sometimes combined with water pumps for furnishing the condensing water to the condenser.

Some vacuum pumps use a slide valve for the pump cylinder, as was shown in Fig. 65; others use a rotary valve accomplishing the same object as a slide valve, and still others use a number of small regular pump valves automatically opened and closed by the flow of the fluid pumped. To prevent entrance of air through the piston-rod stuffingbox of a vacuum pump it is common practice to form a water seal at that point, which consists sometimes of an annular space around the piston rod that is kept continually filled with water. In other cases a trough is formed around the outside of the stuffingbox, which is kept filled with water, thus submerging the stuffingbox.

Wet-vacuum pumps applied to condensers remove from the bottom of the condenser the condensed steam, which is usually called the *condensate*, as well as water vapor and air; if a very high vacuum is desired a dry-vacuum pump is sometimes connected to the top of the condenser in addition to the wet-vacuum pump connected at the bottom.

TURBO VACUUM PUMP

104. A *turbo vacuum pump*, also called a *turbo air pump*, is a form of pump having a rotating impeller from which water is ejected radially at very high speed, this water entrapping air or vapor admitted around the circumference of the impeller, which air or vapor mingled with the water leaving the impeller is discharged from the machine. Turbo air pumps are applied to condensers in steam turbine work, or to the top of vessels in which a very high vacuum must be maintained, and accomplish the same object as reciprocating dry-vacuum pumps.

The principle of a turbo air pump is shown in Fig. 66. Water from some convenient source of supply enters at the center of the impeller *a*, which is rotated at a high speed by a steam turbine or by some other convenient source of power. The impeller is of the double side plate type, has a number of ports in it, and rotates in a direction opposite to that of a centrifugal pump. Consequently, water passing through the ports instead of being discharged quietly from the periphery of the impeller is violently hurled from it in a substantially radial direction, passing across an annular air space *b* connected by piping to the vessel in which a vacuum is to be maintained. The many slugs of water hurled across the air space *b* carry air with them into the tapered channels of the stationary compression ring *c*. Owing to the channels in the compression ring *c* decreasing in area toward the periphery, the outwardly moving slugs of water become longer automatically, thus compressing air entrapped between the slugs. The mixture of air and water is discharged from the compression ring into a circular casing, and discharged into some convenient source of disposal, as an open tank, when the water can be taken into the center of the impeller again.

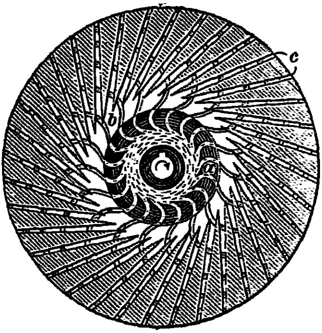


FIG. 66

105. A turbo air pump built by the Wheeler Condenser and Engineering Company, Carteret, New Jersey, is shown in perspective in Fig. 67, with some parts broken away in order to exhibit clearly the construction. The water inlet for the pump is at *a*; this water is delivered to the center of the double side plate impeller *b* which makes a water-tight joint with the side

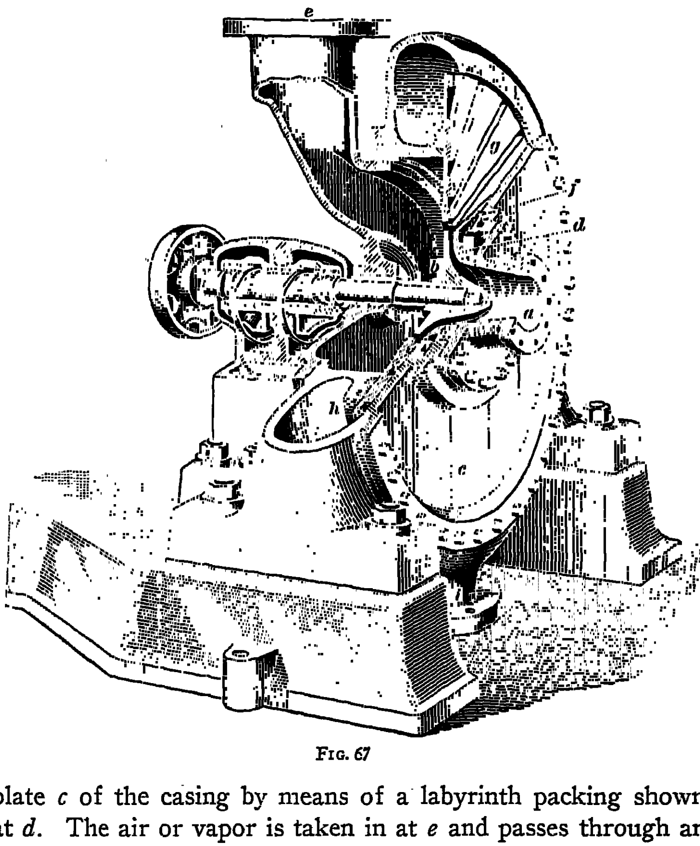


FIG. 67

plate *c* of the casing by means of a labyrinth packing shown at *d*. The air or vapor is taken in at *e* and passes through an annular opening *f* between the periphery of the impeller *b* and the inner circumference of the compression ring *g* to the space into which the impeller hurls water. The mingled water and air passes into the space *h* of the casing and is discharged from the nozzle *i*.

For surface-condenser work a centrifugal hot-well pump and a turbo air pump are often combined into a single casing; the hot-well pump continuously removes the condensate from the bottom of the condenser, and the turbo air pump removes the air and water.

EJECTOR VACUUM PUMP

106. The Worthington hydraulic vacuum pump shown in cross-section in Fig. 68 is an example of an ejector type of pump. Water under considerable pressure enters at *a*, view (*a*), and is discharged through an annular opening at *b* against the so-called jet wheel *c*. This jet wheel is mounted so it can revolve freely; it has a number of ports *d* formed in it, which are inclined so that the impulse of the water entering the jet-wheel ports causes the jet wheel *c* to revolve at high speed, and thereby force the water to leave the lower end of the jet wheel in a succession of slugs. Air or vapor entering at *e* is entrapped between these rapidly downward moving slugs of water and, mixed with the water, is discharged through the pipe *f*. A side view and top view of the jet wheel is shown separately at (*b*).

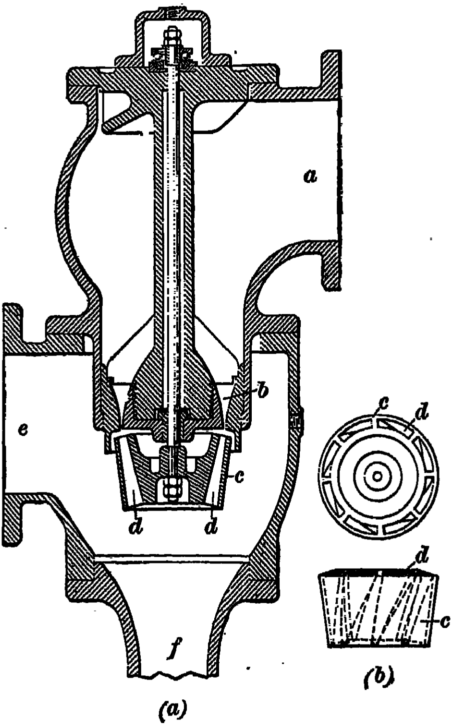


Fig. 68

Water may be supplied to this vacuum pump from a city similar water-service main, or an arrangement as shown Fig. 69 may be employed, where a centrifugal pump *a* tak water from an open tank *b* and discharges it into the vacuu

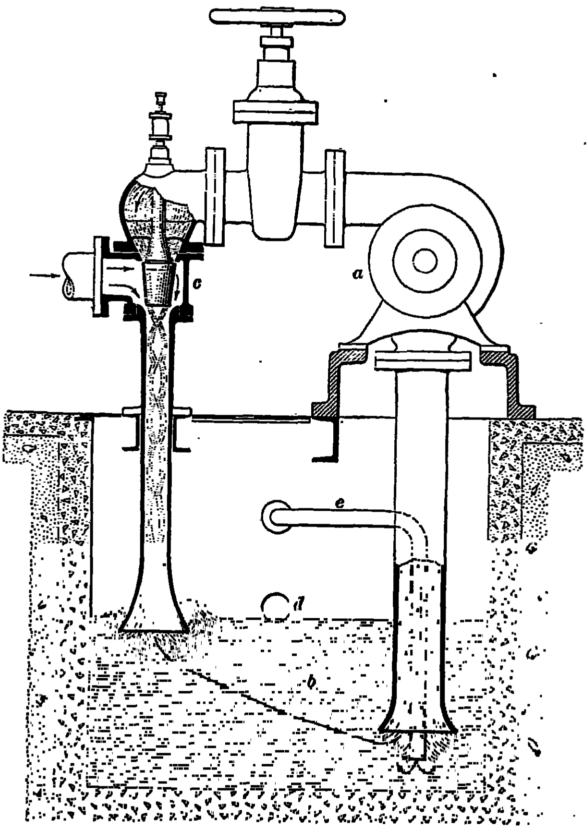


Fig. 69

pump *c*, whence it returns to the tank *b*, so that most of the water is used over and over. An overflow opening *d* prevents the water in the tank from rising too high from any cause; the tank is filled, and any deficiency of water made up, through the make-up pipe *e*.

PUMPS

(PART 2.)

DETAILS OF PUMP WATER ENDS.

PUMP PLUNGERS.

CONSTRUCTION.

1. The smaller sizes of pump plungers are usually made of solid round bars of metal turned smooth, so as to work through a stuffingbox with as little friction and wear as possible. For larger sizes the plungers are frequently of cast iron and are often made hollow to reduce the weight and amount of material required. Incidentally, it may be remarked that a hollow plunger is easier to move than a solid one, all other conditions being equal. This is due to the fact that the water buoys up a hollow plunger more than a solid one. In large horizontal pumps hollow plungers are often so proportioned that they actually float in the water, thus relieving the stuffingboxes of the weight of the plungers and reducing the wear.

2. Fig. 1 shows a simple form of solid plunger pump, such as is often used for feeding boilers. The plunger works through a stuffingbox of the ordinary pattern, packed with hemp or some of the common types of soft piston-rod packing.

3. Fig. 2 shows three styles of large, hollow, cast-iron

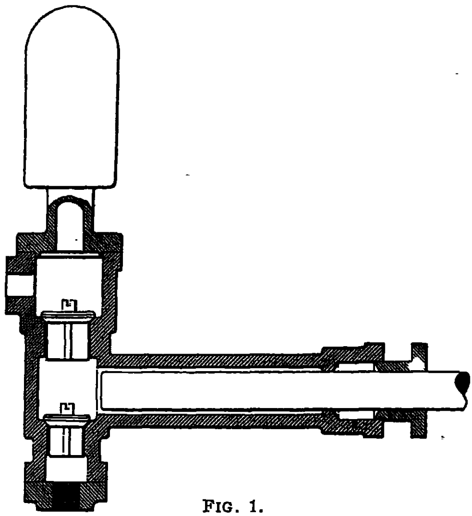


FIG. 1.

plungers, with methods of attaching them to the pump rods. The packing for these plungers, when used for moderate pressures, is usually hemp contained in a stuffingbox of the ordinary pattern.

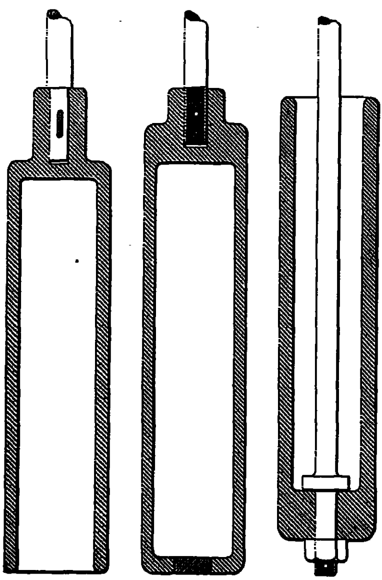


FIG. 2.

PLUNGER PACKING.

4. When the pressure under which the pump works is very heavy, **U**-shaped leather packing is sometimes used. Fig. 3 shows three methods of holding these **cup leathers,** as they are called. The section at (*b*) shows the leather *o* held in a recess cast in the upper end of the

pump cylinder. In this case it is necessary to remove the plunger *D* in order to insert a new leather or to examine an old one. Experience also shows that the leather bears against the plunger with the greatest force at the bend *B* and fails at that point first. In (*c*) the leather is held in its recess by a gland *s*, and is also supported by a brass ring *C*,

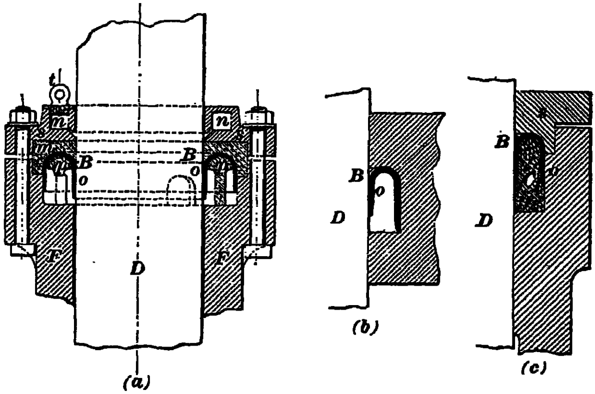


FIG. 3.

which prevents the severe pressure of the leather against the plunger at *B*. A more elaborate packing is shown at (*a*); the gland *s* is lined with a brass ring *m*, which holds the leather *o* down on a brass supporting ring *p*. A chamber *n* in the gland serves to hold oil for lubricating the plunger.

The form of packing shown at (*b*) is cheap, but in addition to the difficulty of inserting the leather, it is difficult to cast the recess so that it will fit the leather properly. In either of the forms shown in (*a*) and (*c*), the gland can be accurately turned to bear against the curved portion of the leather, thus forming a better support and increasing the life of the packing.

5. Fig. 4 shows an inside-packed plunger with a removable stuffingbox designed for hemp packing. This construction is better than merely providing a close-fitting bushing, especially when the water is gritty and thus liable to wear the plunger.

Inside-packed plunger pumps have several disadvantages. When the packing becomes worn, the heads of the pump cylinder must be removed in order to tighten or renew it, and

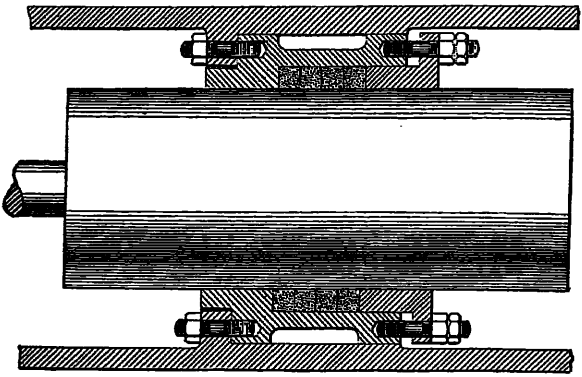


FIG. 4.

besides, there is no way of detecting leakage when the pump is working. With gritty water, especially when working under high pressures, these disadvantages become serious.

6. Fig. 5 shows a good arrangement of plunger, stuffing-box, and gland. This type of plunger and stuffingbox is

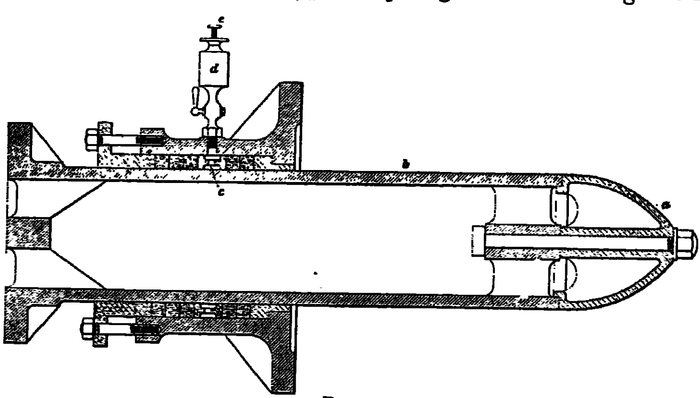


FIG. 5.

much used in mining pumps. The plunger cap *a* is made of acid-resisting metal, while the plunger *b* proper is made of cast iron, it having been found in mining work that the

plunger cap or point is the only part that is attacked by acid water. Apparently the play of the plunger through the stuffingbox and grease prevents the water attacking its surface. An improved form of **grease ring** is shown at *c*. This ring fits into the stuffingbox and is placed between the rings of fibrous packing. It is recessed both inside and outside and has several holes by which the outside recesses connect with the inside recesses. The outside recess is in connection with the grease cup *d*, which is provided with a cock. When it is desired to grease the plunger, the cock is opened and the grease forced in the space around the grease ring by the screw *e* on top of the grease cup. This is done once or twice during the day, and the cock is then closed so as to relieve the grease cup of the water pressure and to prevent consequent leakage. The stuffingbox is bolted directly to the pump chamber, which may be of any type, but for high-pressure mine work it is generally circular. This type of plunger and stuffingbox has been used with much success in the anthracite coal regions.

PUMP PISTONS.

7. Pistons for force pumps are made in a variety of forms. Fig. 6 shows a piston with fibrous packing held in place by a follower. The follower is fastened to the piston by means of an extension of the piston rod beyond the nut that holds the piston in place.

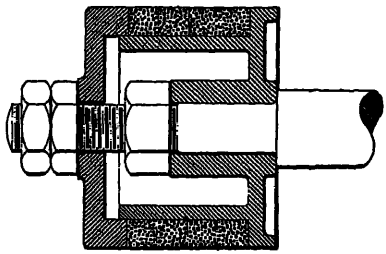


FIG. 6.

8. An excellent packing for small pistons is shown in Fig. 7. It consists of a metallic piston made up in three parts, between which are clamped two cup leathers, as shown.

9. Pistons for suction and lift pumps must be provided with valves that allow free passage for the water through

the piston in one direction and prevent its return. These

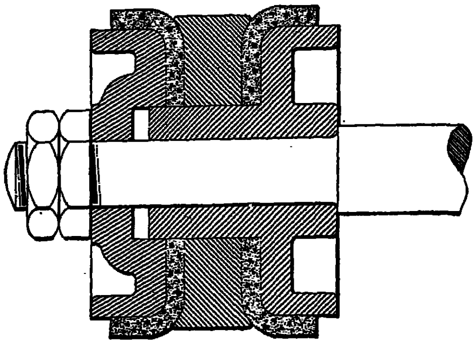


FIG. 7.

valves may be of any design that will furnish the required area of passage and at the same time will be strong enough to withstand the pressure of the water.

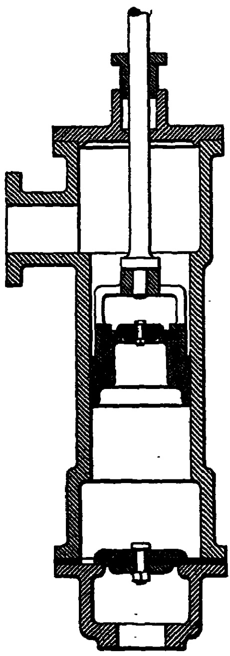


FIG. 8.

10. For small pumps and moderate lifts, leather **clack valves,** Fig. 8, are often used. They consist simply of a leather disk held at one side and strengthened by a metal plate on top. The leather when wet forms an excellent hinge and a tight valve. Leather clack valves are also used for the suction and delivery.

11. For lift pumps working under high pressures, the valves shown in Fig. 9 give good results. The piston shown at (*a*) has a rubber disk valve working on a gridiron seat. The valve is guided by a central spindle *s* and is held on its seat by a light helical spring that acts on a plate on top of the rubber disk. This piston is very long and has no separate packing.

12. The valve shown at (*b*) is for very heavy pressures. It consists of a metal disk guided by a central spindle *s* and held down by a helical spring in the same manner as the

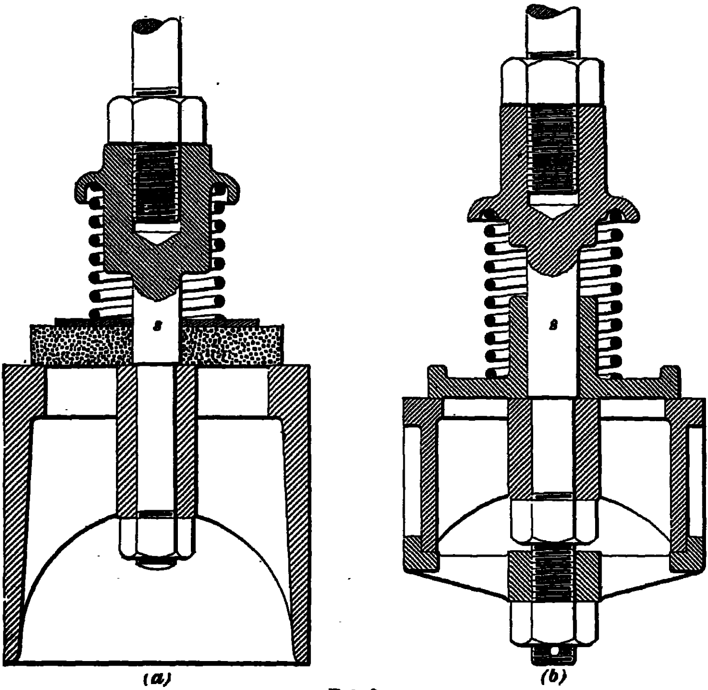


FIG. 9.

rubber valve. The piston is made with a follower plate for the purpose of holding a fibrous packing in the same manner as the piston shown in Fig. 6.

PUMP VALVES.

REQUIREMENTS.

13. The most important details of a pump of any kind are the valves. They must be so designed and constructed that they will fulfil all the following conditions as thoroughly as possible:

(*a*) They must open freely under a light pressure.

(*b*) The net area of the passages through the valves should be great enough to limit the velocity of flow through them to 240 feet per minute.

(*c*) The lift of the valves should be small.

(*d*) The passages for the water should be as direct as possible.

(*e*) The valves must close tightly under all conditions.

(*f*) The valves and their seats must be durable and of such materials as are not easily affected by the impurities in the water.

(*g*) The valves must return to their seats quickly and without shock as soon as the current through them is stopped.

(*h*) The valves and seats must be easily repaired or removed when worn.

A great variety of valves have been designed with a view of satisfying these requirements, taking into consideration the widely varying conditions under which pumps must work.

CONSTRUCTION.

14. Disk Valves.—Fig. 10 shows two valves of a type much used in all classes of pumps for ordinary pressures and service. The valve *v* consists of a vulcanized India-rubber disk that rests on a gun-metal or brass seat *s*. The seat is threaded at *t*, so that it can be screwed into the deck of the valve chamber and thus can be easily removed. The part of the pump chamber that contains the valves is usually called the **valve deck**,

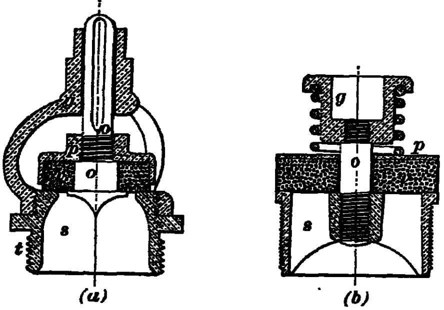


FIG. 10.

and it is spoken of as the **suction valve deck** and **delivery valve deck** in accordance with the kind of valves it carries. In the design shown at (*a*), the valve is fastened to a spindle *o* by a cap *p*. The spindle is guided by a cage-shaped guard *g* screwed on to the valve seat. The lower end of the spindle is made conical, so as to change the direction of motion of the water gradually and to reduce the resistance to flow. In the design shown at (*b*), the spindle *o* is screwed into the valve seat and carries a guard *g*. A helical spring between this guard and the plate *p* helps to seat the valve quickly.

The size of these valves varies from 2 to 6 inches in diameter, the most common size for ordinary conditions being 3 inches.

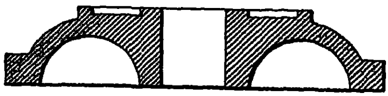


FIG. 11.

15. When used for pumping *hot* water, the disk must be made of a composition that will not be affected by the heat and for very high pressures metal disks are used, generally of the form shown in Fig. 11.

16. Fig. 12 shows the construction of a large disk valve, such as is often used in mine pumps. The valve seat *A* is held in place by the flange *B* and is perforated, as shown in the top view of the seat, by a large number of small holes. The valve *C* is made of soft rubber and is placed within the bronze or composition cap *D*. The head of the bolt *E* forms a stop and the spring *S* assists the valve in closing.

17. Clack Valves.—A section of a clack valve is shown in Fig. 13. The **clacks** *A* and *B* are lined with leather on the bottom so as to make a tight fit on the seat without having to do much fitting. A stop *C* prevents the valves opening too far, while *E* is the pin on which the clacks are hinged. A cylindrical casing *D* forms the valve seat; it may be easily renewed when worn. These valves are of the type known as the **butterfly valve,** and are much used for pit pumps at mines on account of their cheapness and simplicity of construction.

18. Single-Seat and Double-Seat Valves.—A **single-seat valve** that is suitable for high pressures, up to heads of 500 feet, is shown in Fig. 14, where *A* is the valve; *B* is

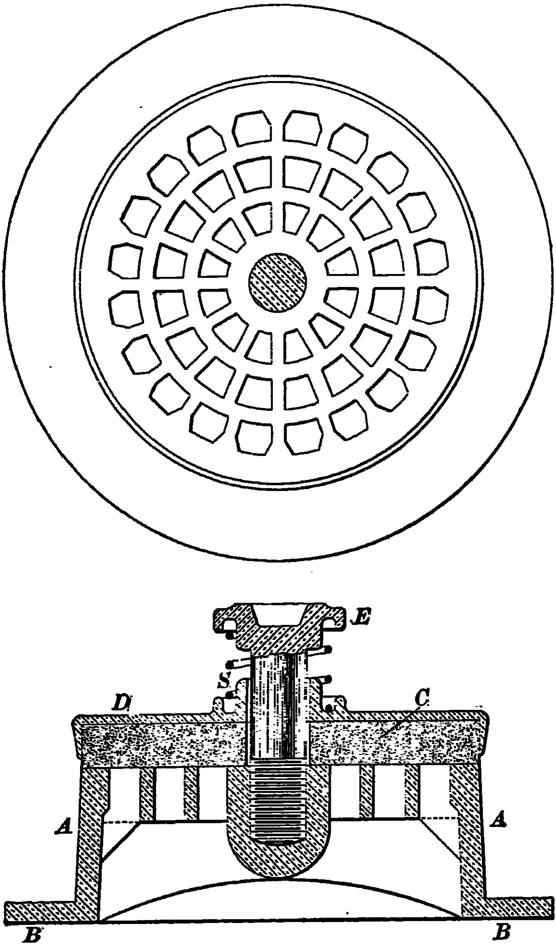


FIG. 12.

a stem solid with the valve that acts as a guide inside the bearing *D*; and *C*, *C*, *C*, *C* are rubber rings which are kept in position by means of the stem and are separated by the

washers E, E, E. These rings prevent shock as the valve lifts and also help to close it quickly, thus serving the same purpose as the helical spring in Fig. 10 (*b*).

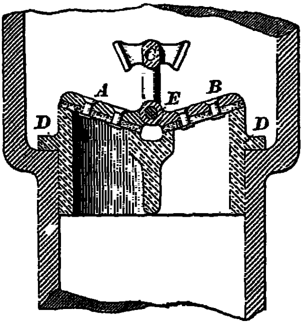


FIG. 13.

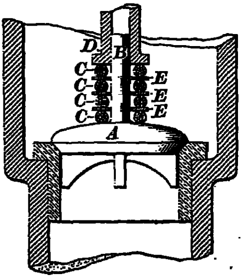


FIG. 14.

19. A section of a **Cornish double-seat valve** is shown in Fig. 15. This valve gives excellent results when used in

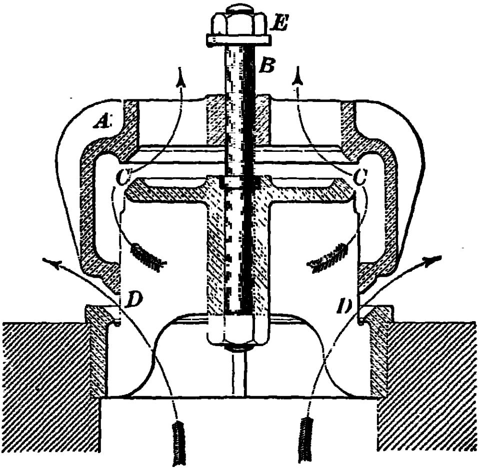
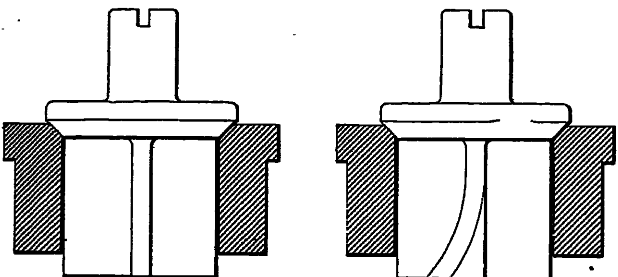


FIG. 15.

large pumps working under high pressures and has been applied to pumps working under heads up to 700 feet. It is called a *double-seat* valve because it has two seats and two

openings for discharge. The casing A slides on the vertical stem B, its lift being regulated by the nut and washer E; when down, it rests on the valve seats C and D. When the pressure below becomes greater than that above, it raises the casing, and the water is discharged through the circular openings at C and D. The rib around the outside of the casing is for the purpose of strengthening it. The valve seats are conical. The figure shows that one opening discharges the water under the lower edge of the valve and the other through the inside.

20. Wing Valves.—The **wing valve** shown in Fig. 16 (*a*) is largely used in power pumps for feeding boilers and in



(*a*) (*b*)

FIG. 16.

hydraulic pumps for high pressures. The valve and seat are made either of hard brass or of gun metal and are ground together to secure tight closing. The lower portion of the wings is sometimes curved as shown at (*b*), the object being to give the valve a partial rotation at each stroke of the pump. This compels it to seat at a new place with each stroke and tends to wear the valve and seat more evenly.

21. Pot Valves.—Fig. 17 (*a*) is a sectional view of a **pot valve.** This type of valve is used principally on mining pumps for lifts up to 1,000 feet. They are made separate from the pump chambers and may be readily replaced when broken or worn. The cover *a* is secured by hinged bolts, so that it may be quickly removed for access to the valve *b* and the valve seat *c*, which is made of composition and pinched

between the pot and the pump chambers. The valve spring *d* surrounds the valve guide *e*.

22. Fig. 17 (*b*) shows a type of pot valve used for high lifts up to 1,200 feet. The valves are made small and faced with hard rubber; a group of them is placed in one heavy

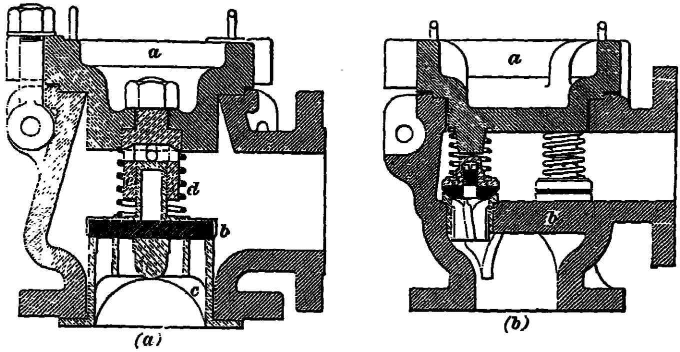


FIG. 17.

pot which is bolted to the pump chamber. Access to the valves may be had by removing the cover *a*. The valve seats are made of composition bushings forced into the valve deck *b*.

AIR CHAMBERS.

PURPOSE.

23. Even in double-acting pumps there is an interruption of the flow at the end of the stroke, when the piston changes its direction of motion. This has the effect of bringing the column of water in the suction and discharge pipes to rest at the end of each stroke, and this column of water must be set in motion again as the next stroke is made. If the pipes are long, the force required to stop and start the water will be very great, and there will be a severe shock at the end of every stroke that will absorb power and subject the pump and pipes to great stresses.

This difficulty is removed and the flow through the pipes is made more continuous and steady by the use of **air chambers.** An air chamber is a vessel containing air and is attached either to the pump just outside of the discharge valves or to the discharge pipe near the pump. While small duplex pumps are often run without an air chamber, it is better in general to fit one to all pumps, since its effect will always be beneficial.

DELIVERY AIR CHAMBERS.

24. Principle of Action.—Fig. 18, which shows an air chamber attached to the discharge pipe of a single-acting

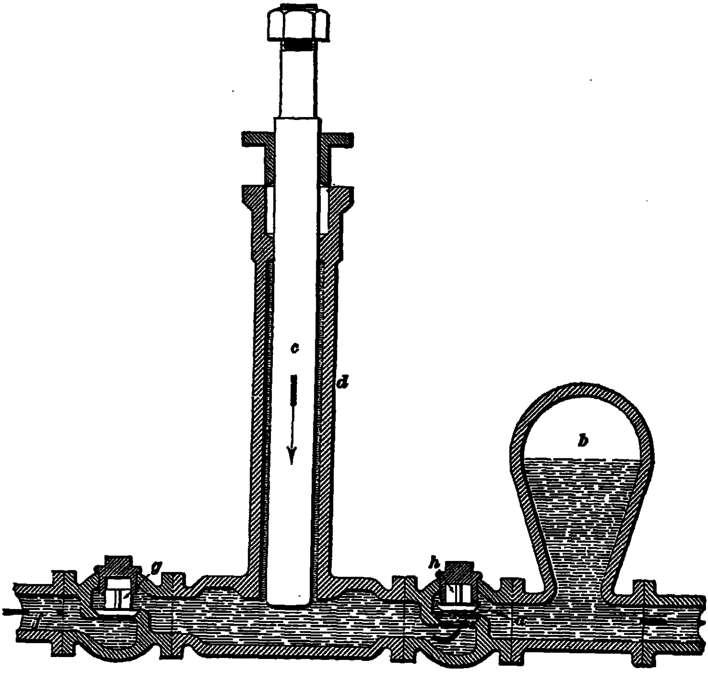


FIG. 18.

plunger pump *d* for boiler feeding, will illustrate the principle of action of an air chamber. The water, after being

drawn in through the pipe *f* past the valve *g*, is forced by the plunger *c* past the valve *h* into the discharge pipe *a*, part of it flowing into the air chamber *b* and compressing the air therein. When the plunger reaches the end of its stroke and no more water is being forced into the discharge pipe, the compressed air in the air chamber forces the extra water out through the discharge pipe. In this way the air chamber acts as a *reservoir* that receives its supply during the inward motion of the plunger and gives it out again in a nearly steady stream. The air in the air chamber acts as a spring that absorbs the extra force during the inward stroke of the plunger and gives it out during the return stroke, thus relieving the pump and pipe of shocks and providing a nearly constant rate of flow from the discharge.

25. Size of Delivery Air Chamber.—The proper size of an air chamber depends on the type of pump, the speed at which it works, the length of the discharge pipe, and the pressure head against which the pump works. For ordinary double-acting pumps working against moderate pressures and at ordinary speeds, the cubical contents of the air chamber should be not less than 3 times the piston displacement. For pressures of 100 pounds per square inch and upwards or for high piston speeds (as in the case of fire pumps), the capacity of the air chamber should be at least 6 times the volume of the piston displacement for a single stroke.

26. Loss of Air.—Under the increased pressure in the air chamber, the air is absorbed by the water and gradually passes off with it. In this way all the air will finally pass off and the chamber will be made useless if no means are provided for renewing the supply.

27. A simple device for maintaining the supply of air in the air chamber of large pumps is shown in Fig. 19. A piece of 2½-inch wrought-iron pipe *c* about 30 inches long is connected to the end of the pump cylinder *a* in a vertical

position, by means of a gate valve *b*, or cock. A $2\frac{1}{2}$-inch **T** *d* at the upper end of this pipe is connected at one end of the run with a $1\frac{1}{4}$-inch check-valve *e* opening inwards, and at the other end with a $\frac{3}{4}$-inch check-valve *f* that opens outwards. The valve *f* is connected with the air chamber through the pipe *g*.

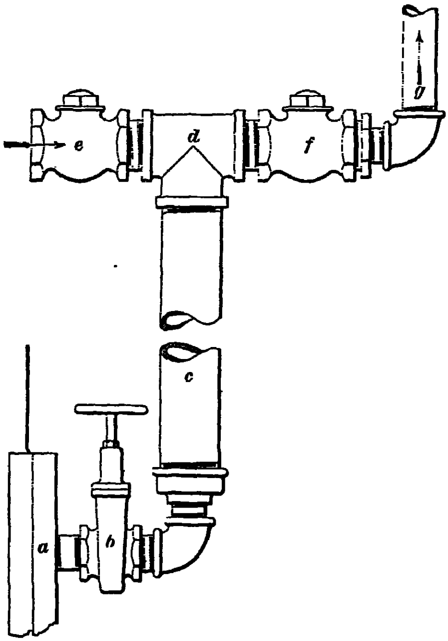


FIG. 19.

This air pump is operated as follows: When the pump is working, open the valve *b* to fill the pipe *c* with water; then partially close *b* until the check-valves *e* and *f* begin to work. This is easily determined by the click of the valves when seating. Its working may be described thus: When the valve *b* is opened, water fills the pipe *c* from the pump cylinder *a* during the discharge stroke of the pump. By partly closing *b* when *c* is full, the pump during the suction stroke will draw a part of the water from *c*, and air will flow in through *e* to take its place. During the next discharge stroke of the pump, more water is forced into *c*, driving the air out through *f* and *g* into the air chamber. If *b* is opened too wide, all the water will be drawn out of *c* during the suction stroke and air will be drawn into the pump cylinder from *e*; but by properly regulating the opening, a column of water is kept in *c*, which acts as a piston that moves with the strokes of the pump and pumps air into the air chamber.

28. Alleviator.—When pumps work under pressures greater than that due to a 350-foot lift, air chambers are not of very much service, owing to the fact that the air escapes from the air chambers either through the pores of the iron or at the joints, or it is absorbed and carried off by the water; in such a condition an air chamber gives the pump no relief whatever. To obviate this defect **alleviators** are used. An alleviator is shown in Fig. 20. It consists of a plunger *a* working through a water-packed stuffingbox. On top of the plunger are arranged springs that may be in the form of rubber buffers or helical coil springs. In the type shown rubber buffers *b*, *b* are used, which are confined by the tie-rods *c*, *c*, the yoke *d*, and the plates *e*, *e*. When the pressure in the pipe exceeds the working pressure, the plunger *a* is forced out through the stuffingbox and relieves the pump of the shocks that would otherwise occur. Alleviators may be placed anywhere on the delivery pipe, but are preferably placed in such a position that the direction of the moving water is in line with the plunger *a*.

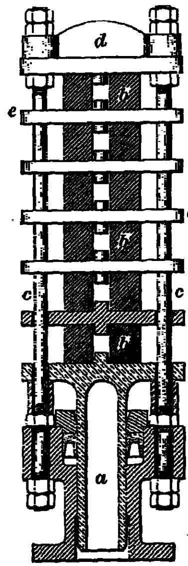


FIG. 20.

SUCTION AIR CHAMBERS.

29. Purpose.—With a long suction pipe or a pipe with numerous bends and valves, the resistance to the flow of the water through it will be considerable, and a great deal of force will be required to start and stop the water in it with each stroke of the pump. In some cases the force required is so great that the pressure of the atmosphere is not sufficient to set the column of water in motion quickly enough to fill the pump chamber as fast as the piston moves. This makes the action of the pump imperfect and causes a

severe blow, called the **water hammer,** when the piston again meets the inflowing water.

30. The difficulty mentioned in Art. **29** can best be remedied by the use of a chamber, called a **vacuum chamber** or a **suction air chamber,** attached to the suction pipe as near the pump as possible. In its general form a vacuum chamber resembles an air chamber, but the pressure in it instead of being greater is always less than the atmospheric pressure. When the pump is drawing water, the air in the vacuum chamber expands and forces the water below it into the pump; at the same time the pressure of the atmosphere forces water in through the suction pipe to balance the reduced pressure in the vacuum chamber. The vacuum chamber is again partly filled and the air in it is compressed during the discharge stroke of the pump. It thus acts as a reservoir that receives from the suction pipe a nearly steady supply, which is given up intermittently to the pump.

31. Special Form of Suction Air Chamber.—Fig. 21 shows a special form of a suction air chamber in diagrammatic form. The suction pipe *a* connects to a suitable chamber *b*, which has a tube *c* projecting downwards to

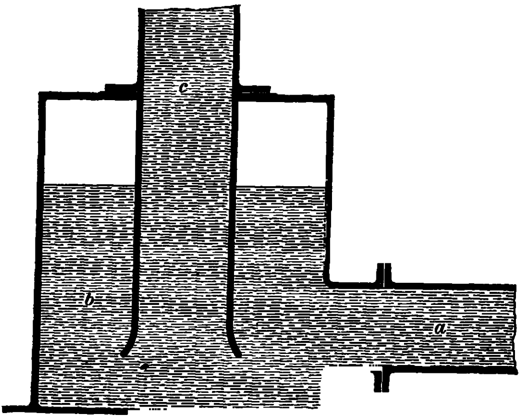


FIG. 21.

within a short distance of the bottom. The tube *c*, which is called a **draft tube,** connects to the pump chamber. When water first flows into the chamber *b*, it entraps some of the air as soon as the water seals the bottom of the draft tube; this air is then compressed while the water flows up the draft tube, and by its expansion and compression permits a steady flow in the suction pipe.

32. Size of Vacuum Chambers.—For ordinary cases, the vacuum chamber may be made half the size of an air chamber working under the same conditions. A good rule is to make the cubic capacity of the vacuum chamber for a single pump twice that of the displacement of the piston for a single stroke.

33. Location.—Suction and delivery air chambers should, if possible, be placed at a bend in the pipe and close to the pump and in such a position as to be in line with the flow of water in the pipe. If placed at right angles to the flow of water, as in Fig. 18, their efficiency is somewhat impaired. Both suction and delivery air chambers should be provided with glass water gauges so that the height of the water can be determined at a glance. It is not customary to provide the air chambers of small pumps with water gauges.

PUMP FOUNDATIONS.

GENERAL CONSIDERATIONS.

34. The foundation for pumping machinery depends entirely on the type of pump. Generally speaking, much less foundation is required than for steam engines occupying about the same space. Direct-acting duplex pumps probably require the least foundation of any kind of steam pump, for here the piston and plunger motion is almost opposite and the balancing of the machine in line with the plunger motion is complete, and the strains due to reversing are contained almost wholly within the machine itself. Small duplex-pump foundations are made of a solid mass of brick or

concrete, while large pumps are often set on separate piers, one for the water ends and one for each pair of steam ends in case of a duplex, compound, or triple-expansion engine. Of course, the foundations must go down to sufficiently hard soil to bear up the weight of the pump, or if the soil be loose sand or gravel, the foundation must be spread out sufficiently to insure the pressure not exceeding, say, 1 ton per square foot. The foundation should go deep enough to allow the surrounding soil sufficient hold upon it to keep it firm and steady. The minimum depth for a small pump should not be less than 2 feet. Single-cylinder pumps require a somewhat heavier foundation than duplex pumps, owing to the greater shocks to which they are subjected.

35. Crank-and-flywheel pumps require considerably more foundation than direct-acting machines, on account of the much higher speeds possible and the weight and lack of balance of the reciprocating parts. Crank-and-flywheel pumps of the controlled-valve type, as the Riedler pumps, which usually run at a high speed, require foundations fully as heavy as those for steam engines of equal size.

MATERIAL AND FOUNDATION BOLTS.

36. Foundations should be built of hard brick laid in cement mortar, concrete, or, in the case of large pumps, of stone, if it can be readily secured. All pumps should be held down by foundation bolts. In the case of small pumps the bolts are provided with a steel or wrought-iron plate washer built solidly into the foundation, while large pumps have tunnels or pockets for access to the lower foundation washer and nut. If the foundation bolts are built in solid, box washers should be used.

FOUNDATIONS FOR LARGE PUMPS.

37. In the case of large vertical pumping engines, the masonry required to form the pump pit and to support the superstructure is of ample mass for all foundation purposes; in fact, large arched chambers and tunnels are often used to

save foundation materials in this class of pumping engine. These large pumping engines are often located at or near a water supply where the soil has not sufficient rigidity to support the weight. In this case piling must be resorted to, on which the foundation proper is constructed.

USE OF FOUNDATION TEMPLET.

38. A foundation templet should always be used in which the foundation-bolt holes are carefully laid off, preferably from the actual castings, and the various heights of bosses or thicknesses of casting through which the bolts pass are marked. The templet should be carefully set with reference to the suction and delivery connections, so that when the pump is set up, the fittings and pipes will connect up properly. In large pumps it is customary to arrange the pipe connections in such a way that a short space is left between the piping and the pump. This space is then measured after the pump and piping are in place, and a distance piece is made to suit the measurement and then put in place.

FOUNDATIONS FOR SMALL PUMPS.

39. Small pumps of the single-cylinder and duplex type are usually provided with two points of support only, one of which is rigidly bolted to the foundation, while the other is left free. This prevents the pump being thrown out of line, if properly constructed originally. When both the steam and water ends are bolted down, care must be taken not to twist or throw the pump out of line. In making the steam and water connections, the pipes should come fair to their connections and should not be sprung into place. Stresses on the pump structure due to winding foundation surfaces and sprung pipe connections should be guarded against, particularly with steam-thrown valves, as these are very sensitive and must be perfectly free. Any slight springing of the valve chamber will bind the valve and prevent its operating.

PIPING.

SUCTION PIPING.

40. Location of Pump in Respect to Supply.—Before a pump can be properly located, the location of the source of supply of the liquid to be pumped must be taken into consideration. Since the atmospheric pressure of 14.7 pounds to the square inch will balance a column of water 34 feet high, it is evident that with that atmospheric pressure the pump must not be placed more than 34 feet vertically above the surface of the water to be pumped. But since a perfect vacuum cannot be obtained by mechanical means, and since the flow of the water is retarded by friction in the pipes and passages, the limit of vertical lift by atmospheric pressure is reduced to about 28 feet at sea level in actual practice. The actual lift, precisely as the theoretical lift, varies with the atmospheric pressure, and hence will become smaller with an increase of altitude above sea level, since the air becomes lighter and its pressure less.

41. Run of Suction Pipe.—The pump should be placed as near the source of water to be pumped as is possible, both vertically and horizontally. The suction pipe should be as straight as possible; if bends are necessary, they should be made by bending the pipe to a long radius or by using long-turn fittings. The suction pipe should be one diameter from end to end; all enlargements or reductions in size tend to disturb the uniform flow of the water so essential to a proper filling of the pump chamber. If from necessity the suction pipe is very long, it will be well to increase the size somewhat; the reduction at the pump chamber should then be made by a long conical fitting. For ordinary service pumps the diameter of the suction pipe should be such that the velocity does not exceed 200 feet per minute, assuming that the flow of water is constant. If the vertical lift be high, a suction air chamber should be provided; this will

add much to the uniformity of the pump supply. A foot-valve should also be provided when the lift is high.

42. Foot-Valves.—A foot-valve is a check-valve placed at the lower end of the suction pipe below the water level in the source of supply and opening towards the pump. Its purpose is to prevent the suction pipe emptying while the pump is at rest and to prevent the water in the suction pipe slipping back while running. When the water flows to the pump by gravity, a foot-valve is superfluous; but when the water is lifted by suction it is often fitted, since it will insure a prompt starting of the pump, providing that it is tight enough to hold the water in the suction pipe. In very cold weather and in exposed locations, the foot-valve constitutes an element of danger when the pump is out of use, since it prevents the emptying of the suction pipe. The water in the latter may freeze and burst the pipe. To prevent this, a drain may advantageously be fitted to the lower end of the suction pipe, which is used in cold weather to empty the pipe if the pump is to stand idle for a long time.

43. When foot-valves are used, a relief valve may advantageously be placed on the suction pipe. Generally, the suction pipe is made considerably lighter than other parts of the pump, and if the suction valves should leak when the pump is standing or if the priming pipe be left open, the full pressure of the delivery water will come on the suction pipe and foot-valve, which are not usually designed to withstand such pressures. The relief valve, which should be set to relieve the pipe at a pressure well within its safe strength, prevents overstraining of the suction pipe from this cause. Foot-valves should be chosen with the greatest care; they should be simple and, preferably, of the weighted-lift type or clack valve, and should have at least 50 per cent. excess of area over the suction pipe.

44. Settling Chamber.—If the water to be pumped is gritty or contains foreign substances, a settling chamber is sometimes used, especially when pumping water holding but

a small quantity of sand in suspension. This consists of an iron box conveniently arranged in a horizontal pipe. It is usually of large relative capacity, a settling chamber for a 2-inch pipe being 2 feet × 2 feet × 3 feet long. The pipes enter and leave from opposite sides and near the top. The increased volume of the large box allows the water to move very slowly across the box, giving the suspended sand time to settle to the bottom. The settling chamber should have a removable cover for the purpose of removing the settlings. This device is used on small pumps working on artesian wells.

45. Suction Basket and Strainer.—More universal arrangements for keeping back foreign matter from the working barrel of the pump are the **suction basket** and the **strainer.** The suction basket is usually placed on the bottom of the suction pipe and consists of a box variously shaped and perforated with strainer holes or provided with screens. The suction basket so placed is being replaced by a different form of strainer, which consists of a chamber placed in the suction pipe, located in an accessible position and provided with strainer plates so made that they can be readily removed for cleaning. This strainer is sometimes connected directly to the pump, but it should not be so placed that it will interfere with removing the water-cylinder heads. A short piece of pipe between the strainer and pump nozzle will avoid this interference. The objection to the suction basket on the bottom of the suction pipe is its inaccessibility for cleaning and inspection, a feature that is overcome by the strainer.

DELIVERY PIPING.

46. Run and Valves.—While the suction pipe is very important and must be most carefully laid out and has much to do with the location of the pump, the delivery pipe should not be neglected. A careful adjustment between the supply and delivery pipes should be made in order to produce the

best effect of the whole plant. The delivery pipe should as far as possible be a plain, straight pipe from pump to terminal; when bends are necessary, they should be by as long sweeps as possible. A gate valve or check-valve should be placed near the pump. The check-valve serves the double purpose of relieving the pump of pressure when starting up, allowing it to take hold of the water more quickly, and also holds the water back from the pump when inspection and repairs to the water end are necessary. If a check-valve is not used, a gate valve should be placed at or near the pump delivery to hold back the water in case of repairs to the pump end or accident. This valve should always be a straightway gate valve giving the full clear opening of the pipe.

47. Velocity of Flow.—The velocity of the water flowing through the delivery pipe for direct-acting pumps should not much exceed 330 feet per minute, while for large crank-and-flywheel pumping engines the velocity of water in both suction and delivery pipes is about 300 feet per minute. If the suction pipe is made small, the pump will fail to fill and the plunger will strike the incoming water on its return stroke, producing a violent and dangerous shock. If the delivery pipe is made small, the cost of power required to force the water through the pipes at a high velocity will very quickly overrun the interest and depreciation on a larger pipe.

AUXILIARY PIPING.

BY-PASSES.

48. Water-End By-Pass.—**By-pass pipes** are pipe connections from above to below the delivery-valve deck and are of much more use on crank-and-flywheel pumps than on direct-acting machines. In the case of compound pumps, when starting up, the force of the full steam pressure on the high-pressure piston is not sufficient to move the plungers

against the resistance due to the head of the water in the delivery pipe; but by opening the valve (which, by the way, should always be a gate valve) in the by-pass piping, the pressure on the plungers is relieved for a sufficient number of strokes to allow the steam to reach the low-pressure piston, when the combined force of the two pistons will do the work and the by-pass pipe can be closed.

49. By-pass water pipes have another function on crank-and-flywheel pumps. Unless these machines are fitted with very large flywheels, their limit to slow running is often not as low as desired. By opening the valve in the by-pass pipe, part of the water can be returned to the pump chamber and the amount of water actually pumped reduced to any desired quantity permitted by the size of the by-pass. It should not be overlooked that this is accomplished at a very considerable loss of efficiency, because it takes the same power to move the by-pass water as it does to do the actual pumping, comparing equal quantities. By-pass pipes are usually made $2\frac{1}{2}$ per cent. of the plunger area.

50. Steam-End By-Pass.—It is common practice to fit the steam cylinders with by-pass pipes, allowing high-pressure steam to act on the low-pressure piston in starting, but these pipes are usually so small, compared with the diameter of the low-pressure piston, that the by-pass is unable to hold any pressure behind the low-pressure piston when it is moving. By-pass steam pipes have their proper use in warming up the low-pressure cylinder and connections, and in the case of crank-and-flywheel pumps to move the high-pressure crank off the dead center.

PRIMING PIPE.

51. The **priming** or **charging pipe** is a small pipe run from the delivery pipe beyond the check-valve or delivery gate valve to the suction chamber of the pump. It is particularly useful in the case of long suction lifts to fill the working chamber and suction pipe with water, taking up all

clearances and helping the pump to take hold of the water quickly. This pipe may be from $\frac{3}{4}$ of 1 per cent. to 1 per cent. of the area of the plunger; its size is a matter of little importance, but it should be large enough to fill the suction pipe and pump chamber in a reasonable time, which will depend somewhat on the size and design of the pump chamber and the length of suction pipe. A pipe much larger than 1 per cent. of the plunger area will be required in the case of long inclined or horizontal suction pipes.

WASTE DELIVERY PIPE.

52. A **waste delivery** or **starting pipe** that can be led into any convenient place of overflow should be provided so that the pump, in starting, can free itself of air, for it often happens that a pump refuses to lift while the full pressure against which it is expected to work is resting on the delivery valves, for the reason that the air within the pump chamber is not dislodged but only compressed and expanded again by the motion of the plunger. A pump in this condition is said to be **air bound.** It is well in this case to run with the delivery pipe empty until the air is expelled and the water flows into the suction end of the pump. The waste delivery pipe is fitted with a valve and connected to the delivery pipe close to the pump. When the water flows to the pump and is discharged into the delivery pipe, the valve in the waste delivery pipe is to be closed.

AIR DISCHARGE VALVES.

53. When a check-valve is not used in the delivery pipe and the space between the suction and delivery valves is large and the delivery pipe is full of water, the pump will often refuse to start the water in the suction end, owing to compressed air being trapped between the water in the delivery deck and suction valves. Air discharge valves, each composed of a globe valve and a check-valve, may then

be used on each head, the check-valve opening to the atmosphere, thus permitting the escape of air but preventing its entrance when the globe valve is open. The globe valve is closed when the pump is working properly, as shown by water coming from the check-valve.

GENERAL PIPING ARRANGEMENT.

54. Fig. 22 shows a good arrangement of a pump in relation to the water supply and of the pipe connections. The

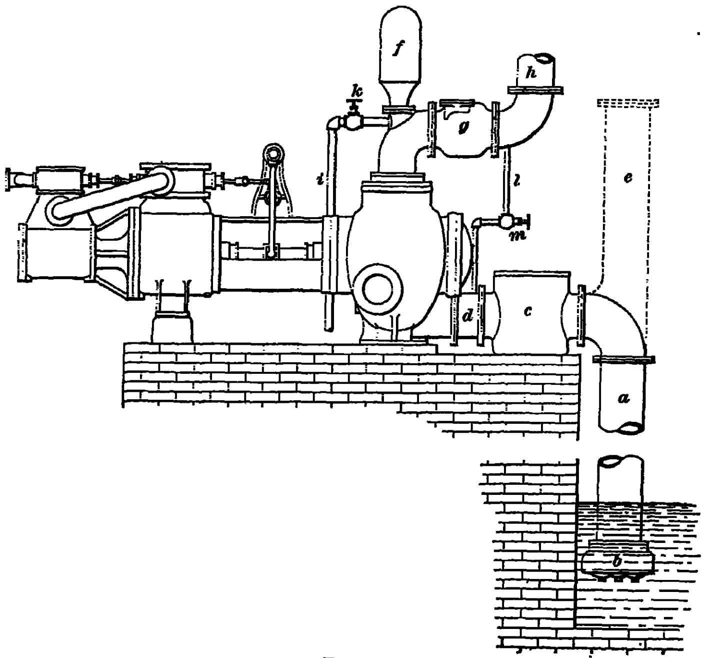


FIG. 22.

suction pipe *a* is fitted with the foot-valve *b* and has a strainer *c* placed close to the pump, from which it is separated by the short distance piece *d*. When the vertical lift is short, say not over 10 feet, and the pump is placed close to the source of supply, a suction air chamber is seldom necessary, but when the lift exceeds 10 feet or when the pump is

at some distance from the water supply, a suction air chamber becomes a necessity. With a vertical suction pipe as shown, the suction air chamber may be made as shown by the dotted lines at *e*. An air chamber *f* is placed on the delivery between the delivery check-valve *g* and the delivery valves. The waste delivery or starting pipe *i* is connected to the delivery between the delivery valves and the delivery check-valve *g*. It is fitted with the valve *k*. The delivery pipe *h* is connected to the suction pipe close to the pump, in this case to the distance piece *d*, by the priming pipe *l*, which is fifted with the stop-valve *m*.

PROVISION FOR DRAINAGE.

55. Proper drain pipes and drain valves should be provided for all parts of the pump, the pipe connections, strainers, etc., in short, for all parts in which water may remain when the pump is not in use and will give trouble by freezing.

Provision for draining the suction valve deck and delivery valve deck is sometimes made by drilling a small hole through the decks; this practice, while simple and cheap, leads to a loss in efficiency, however, since some of the water is constantly flowing back into the suction chamber.

PUMP MANAGEMENT.

INTRODUCTION.

56. If a pump has been properly selected for the service and has been properly designed, built, and erected, it should perform its work without any trouble. All pumps when new are stiff and cranky in their actions, particularly direct-acting pumps. They should be run slowly for a considerable time, and many defects in their action which at first gives rise to alarm will then gradually disappear. Crank-and-flywheel pumps act more smoothly from the start, but do

not come to a proper bearing more quickly or quite as quickly as the direct-acting pump. Crank-and-flywheel pumps usually require considerable skill and study to reduce them to successful working order, as conditions arise that further disturb the lack of harmony between the flywheel and water, and it often taxes the skill of the experienced engineer to make an amicable adjustment between the two opposing forces.

57. Having reduced the pump to satisfactory operation, the attention of the operator should be directed to its maintenance at the least possible expenditure. Each item of expenditure should be separated from the whole and studied independently for the purpose of reducing it to a minimum consistent with the proper maintenance of the plant. The expenditure should at all times be regarded as the item by which interest or dividends are being earned and should not be allowed to become greater.

58. Losses in efficiency arise from wear, from loss of proper adjustments, and from the wrong timing of the various movements that control the distribution of steam, by leakages, by decreased mechanical efficiency due to lack of alinement, by accumulations of foreign matter on and in condenser tubes, suction strainer, and foot-valves, suction and delivery pipes, and in many minor directions. In many plants it is of the utmost importance that they should not be interrupted; it is then the duty of the engineer to predict all possible events that might cause an interruption and have a well-planned line of action prepared so that he may act quickly and with decision to the end of keeping his plant always at work and at the highest efficiency. This plan of action will entail considerable work, study, and, perhaps, some expense in preparation to meet possible contingencies that may never happen; nevertheless, it is well to be ready for any emergency when handling steam machinery and particularly steam pumping engines.

59. In the management of pumps it must be considered that nearly every installation has its peculiarities, some of

which are sometimes not discovered until after the machine is put in service and then perhaps require expensive additions and alterations to meet them. An exhaustive study of existing conditions and resultant conditions when the pump starts to work cannot be too strongly urged.

STARTING PUMPS.

IMPOSSIBILITY OF SPECIFIC RULES.

60. Pumps differ so much in their construction and design that it is entirely impossible to lay down specific rules that will be applicable to every pump. For this reason only *general* rules are here given, which must be modified by the pump attendant to suit every specific case.

GETTING A PUMP READY.

61. Getting Up Steam.—Considering a new steam pump, after it has been properly erected on a suitable foundation and all the pipe connections have been made, the first step in starting the pump is to get up steam in the boiler or boilers in the same manner as is done with boilers supplying steam for any other purpose.

62. Since the boilers are generally in charge of the same person that attends the pump, the general treatment of the pump and the boilers, while steam is being raised, will be considered together. After the steam piping is in place, but before it is finally connected to the pump, all valves in it should be opened wide; while steam is being raised the pistons and valves should be removed from the steam end of the pump so that there is a clear passage for the steam from the boiler to the exhaust after the steam pipe has been connected to the pump.

63. Blowing Out the Steam Piping.—The fires should be started very slowly under the boiler; all the binding

bolts throughout the boiler setting should be perfectly loose and free. If this precaution is neglected, buckstaves or cast-iron fronts will be broken by the expansion of the setting. The guy rods on iron stacks should also be slacked off; in fact, every part that will expand when the plant is started up should be liberated. Before the steaming stage is reached, large volumes of heated air will be driven through the pipes, warming them up gradually. When steam begins to rise, it should be allowed to blow through the piping and valves quite liberally, the object being to clear the piping of sand, grit, and all other foreign matter collected therein during erection. The piping having been blown out thoroughly, steam is shut off and the piping is then connected to the pump.

64. Blowing Out the Cylinder.—When the pressure in the boiler has been raised to the working pressure, the cylinder heads should be put on, still leaving the pistons and valves out of the cylinders. The stuffingboxes should be closed, which is most conveniently done by placing a piece of board between the stuffingbox and the reversed gland and then setting up the nut on the stuffingbox studs. When the gland is drawn home by a nut outside of it, a circular piece of pine board may be placed between the end of the gland and the inside of the nut in order to close the opening through which the piston rod passes. The steam may now be turned on the main steam pipe leading to the pump; by opening the throttle valve wide at short intervals, the sand and scale in the ports and other passages and spaces of the steam end can be blown out. After the cylinders have been blown out, the heads and covers should be removed, and all foreign matter blown into the corners and chambers of the cylinders should be removed by hand. The pistons, valves, cylinder heads, and other covers can now be put in place.

65. The blowing out of the pipes and cylinders after erection is often neglected or but imperfectly done, with

serious consequences to the machine; it cannot be too thoroughly done, particularly in that type of pump where the steam ports and exhaust ports are on top, for in this particular case the sand and grit are deposited in the bottom of the cylinder for the piston to ride upon. If more attention were paid to the thorough cleaning of all steam spaces, we would hear less of cylinders and pistons being cut.

66. Keying Up.—If the pump is of the crank-and-flywheel type, it should be turned a complete revolution by hand to insure that everything clears properly and that no tools or materials used during construction or erection have been left within the machine. The adjustment of all journals, pins, and bearings should then be made. With gib and key ends, it is usual to drive down the key with a soft hammer (lead hammer) until it is home, mark it, drive it back, and then tap it down to within $\frac{1}{8}$ inch of the mark. With wedge ends the wedges usually have an inclination of $1\frac{1}{2}$ inches per foot and the adjusting screw 8 threads per inch. The wedge is drawn up solid and then the adjusting screw is turned back about 20° and locked. Bolted connecting-rod ends are allowed about $\frac{1}{200}$ inch play, using liners and setting the bolts up solid. Main bearings can be adjusted best when the machine is in motion.

67. Packing Rods and Stems. — The packing of all rods and stems is the next step. If fibrous packing is used, the boxes should be filled full and the glands tightened down very moderately. The tightening of the glands can best be done when steam is on and the machine is in motion, when they should be tightened only sufficient to stop leakage and no more. When excessive tightening is required to stop leakage, the packing should be completely renewed. Some pumps are fitted with metallic packings. These packings are usually fitted up by specialists who fully guarantee them, and their directions for use should be carefully followed; in case of failure or unsatisfactory results, the makers should be consulted.

68. Oiling.—The oiling of the machinery is the next step and is a very important one. All rubbing surfaces should be provided with suitable oiling devices appropriate to the particular place and service. The quality of oil should be carefully selected to suit the velocity and pressure of the rubbing surfaces on which it is used. For use within the steam cylinder, heavy mineral oil is the only oil capable of withstanding the high temperature, and in starting up new pumps only, the best quality should be used, regardless of price. A liberal use of this oil for the first month will go far towards reducing subsequent oil bills.

69. The pumping engine, unlike many other types of engines, must often run continuously and without interruption for a month or even longer at a run. This requires that all oiling devices be so arranged that they can be supplied and adjusted while the machine is in motion. It is a good plan to provide two separate sets of oiling systems for all the principal journals, the idea being that if one fails the other can be used while the disabled one is being overhauled. All oil holes should have been filled with wooden plugs, bits of waste twisted in the hole, or some other protection, while the machine was being erected. These should now all be removed and all oil holes and oil channel thoroughly cleaned out. Bearings should be flooded with oil at first to wash out any dust or grit that may have reached the rubbing surfaces.

70. Having turned the machine by hand and inspected all locknuts, setscrews, and clamp screws, the engine may be put under steam. If provided with hand starting gear, this should be used for a sufficient number of turns to make sure that the machine is free from water that may have accumulated in the pipes or clearance spaces. All drain cocks should be wide open when starting and relief valves should be adjusted to blow at the proper pressure. If the engine is condensing, connections from the exhaust port to the condenser should be made absolutely tight. If an independent condenser is used, it should be started before

the main pump is started and a vacuum obtained in advance.

71. So far only the steam end of a large crank-and-flywheel pump has been considered. With the direct-acting single or duplex steam pump, the same general method of procedure should be followed. It may be mentioned here, incidentally, that the direct-acting pump is not so liable to an accident in starting as the crank-and-flywheel pump on account of the absence of kinetic energy stored up in a moving flywheel. This energy when given out by reason of an obstruction in the water end that prevents the free passage of water will greatly increase the pressure, especially when the obstruction occurs near the dead-center positions of the crank. The increased pressure thus produced may easily run up high enough to burst the water end.

72. Using the Dash Relief Valves.—In starting a direct-acting pump when dash relief valves are fitted, they should be closed in order to keep the pistons as far from the heads as possible, for in new installations the unexpected is likely to happen at the water end, and to prevent danger of a breakdown, as in case of a sudden lunge of the pistons, all the margin possible to keep them from striking the heads should be gained.

73. Condition of Water End When Starting.—Assuming that the plungers and plunger rods are packed and the plunger grease cups filled, the water end should be ready to start; if the machine is compound or triple-expansion, the water by-pass valves must be opened until the machine has made a sufficient number of strokes to bring the intermediate and low-pressure cylinders into action, when the by-pass valves should be closed. The suction pipe from the foot-valves to the delivery valve deck must be absolutely tight; anything short of this will cause the water end to refuse to work satisfactorily. All the suction valves and delivery valves should seat fairly and tightly. Care must be taken that there is no obstruction in the delivery pipe, such as a

closed valve, as pumps usually have sufficient margin in the driving force over the resistance to burst the water end particularly if the momentum of a flywheel be added to it.

74. Pressure gauges should always be attached to the suction and delivery pipes, and they should be carefully watched during the process of starting, as trouble at the water end will be promptly recorded by the gauges. The lower end of the suction pipe should be kept well under water, as a slug of air taken into the pump may cause a violent jumping and in a direct-acting pump possibly a striking of the steam pistons against the heads.

75. Watching the Air Chamber.—The delivery air chamber should be carefully watched during the starting and running. This should be provided with a gauge glass showing the height of the water and extent of the pulsation. The air chamber should be charged with air when the air in the chamber is lost, as shown by the rise of the water in the gauge glass. Large pumps are usually supplied with an air charging pump that is attached to and driven by the main pump, or an arrangement of pipes and valves is sometimes improvised for this purpose. In very large pumping plants, an independent air compressor or locomotive air pump is often used for this service. A very good idea of the internal working of the pump can be obtained by placing the ear against the pump chambers; the seating of the valves can then be distinctly heard, and if there is any leak past either the suction or the delivery valves, it, too, is quite audible.

DEFECTS IN PUMPS.

SUCTION-END TROUBLES.

76. The most common causes of pump failures are leaks below the suction valves. These may be at the joints or along the suction pipe or in the pump chamber, and may be due to imperfect connections, leaky chaplets, shifted cores, blowholes, corrosion, or cracks from frost.

77. Small leaks in the suction end which are not sufficient to cause entire failure will cause the piston to jump, i. e., move suddenly, during the first part of the stroke. Leaky valves and plungers reduce the capacity of the pump; if this is the case, they should immediately be refitted and repacked. It is always best to have hot water flow to the pump by gravity; if it is necessary to lift it and the pump works with a jerky action, the lift is too high for the temperature, and one or the other must be reduced. In pumping from wells, care should be taken that the pump is near enough to the water to prevent the water falling below the maximum lift by suction.

78. If the pump pounds soon after the beginning of a stroke, when running fast, it shows that the pump chamber is not filling and that the plunger is striking the incoming water on its return stroke. A suction air chamber will help to remedy the evil. Obstructions under the suction or delivery valves will cause a very decreased output or total failure. A suction strainer or end of suction pipe becoming embedded in sand or clogged with foreign matter will cut off the supply from a pump.

79. Air pockets under the delivery valve deck, caused either by bad design or a shifting core, will very much reduce the capacity and efficiency of a pump. The effect of the air pocket is to entrap air, which is compressed to delivery-water pressure and expands again on the suction stroke. If the relative capacity of the pocket to the plunger displacement is sufficient, the entrapped air will expand to atmospheric pressure, reducing the suction lift to zero; this defect, however small, will always reduce the suction lift and is not easy to remedy; its existence should always be cause for the rejection of a pump.

80. Pounding in pumps is sometimes caused by the water lagging behind the plunger, due to the friction of a small, long, horizontal suction pipe. When suction pipes have a long horizontal run, they should be one or two sizes larger.

DELIVERY-END TROUBLES.

81. Pumps sometimes fail when the full head is resting upon the delivery valves by the air between the suction and delivery valves being expanded and compressed by the motion of the plunger. Air cocks should be provided close up under the delivery decks for discharging the air until the plungers have caught the water. If only a simple cock is fitted, it must be opened during the delivery stroke only and closed shortly before the suction stroke commences. This is to be repeated until a steady stream of water issues from it during the delivery stroke. An automatic air valve, which is simply a small spring loaded valve opening outwardly and closing automatically during the suction stroke, is preferable; this valve should be secured to its seat after a steady stream of water issues during the delivery stroke. Violent jarring and trembling of the pump arises from the delivery air chamber becoming filled with water. It should be recharged with air by means of the air-charging pump, a near-by air compressor, or by a hand air pump.

STEAM-END TROUBLES.

82. The steam end of pumps should not be taken apart needlessly, especially the steam end of direct-acting pumps with steam-thrown valves, as their action is quite complicated, and a very slight misadjustment will cause a failure. If at any time it becomes necessary to dismantle the pump, all the parts, if not already marked, should be plainly marked with steel letters or numbers, rather than with a prick punch or chisel, and suitable gauges, by which all parts can be returned to their correct relative positions, should be made, if this is deemed advisable. In many duplex pumps there are very slight differences in the two sides; for instance, the crossheads that drive the valve levers are not keyed in exactly the same position on the piston rods and the rods are not interchangeable; the pump will not run successfully if they are interchanged. In some pumps with steam-thrown valves, the valve chests are bolted to the cylinders,

and are reversible so far as fitting and bolting goes, but the auxiliary ports are not reversible and will be shut off in both valve chest and cylinder by reversing the chest. In placing the gasket between the valve chest and cylinder of pumps with steam-thrown valves, care should be taken to cut passages through the gasket for the auxiliary ports. The valve levers, pins, and all connections between the piston rod of one side of a duplex pump and the valve of the opposite side should be kept in good condition, as the failure of these parts will cause a serious accident.

83. On duplex pumps the amount of lost motion between the valve stem and the valve should be very carefully adjusted; too little lost motion will cause short stroking, while too much will allow the pistons to strike the heads. If the pistons strike the cylinder heads, the dash relief valves, if fitted, should be closed until the stroke is shortened sufficiently for the pistons to clear the heads. If the stroke becomes too short, the opposite course should be followed. If no dash relief valves are fitted, the lost motion should be made smaller in case the pistons strike the heads.

84. When a compound pump is fitted with a cross exhaust and it is seen that the pump is unable to complete its full stroke, the valve in the cross exhaust should be opened.

TESTING PUMPS FOR LEAKAGE.

85. Testing the Suction Pipe.—Leaks are the most troublesome and most frequent sources of loss of efficiency in pumping machinery. Leaks in the suction pipe or suction system affect the pump most and often cause its complete failure. These leaks can sometimes be detected by the ear, or the flame from a common tallow candle will often locate a leak in the suction by being drawn towards the hole by the air. Sometimes these leaks are very numerous, but so small that any one of them would be difficult to locate and be of small importance; at the same time, their combined effect may be sufficient to seriously affect the working of the

pump. The best way to locate these leaks, which may be at the joints or along the body of the pipe, is to stop up the inlet end of the pipe, uncover it completely, and then put a water pressure on it, say from 40 to 50 pounds per square inch. Any leaks, however small, will then be readily detected. The suction pipe should always be tested for leaks before it is covered, if laid in a trench or otherwise made inaccessible, because it must be made tight before the pump will work successfully.

86. Delivery Pipe Leaks.—Leaks in the delivery pipe, while common and at times more difficult to remedy than leaks in the suction, are plainly evident. They do not affect the action of the pump or its efficiency to any extent, the loss being exactly proportional to the magnitude of the leak.

87. Repairing Leaky Pipes.—Probably the most satisfactory method of procedure in case a leaky section of pipe is discovered is to discard it and replace it with a new one. Circumstances, however, do not always permit this to be done, and then temporary repairs should be made. The manner of making the repair obviously depends on the position and extent of the leak and calls for the exercise of judgment and some skill.

88. Small leaks in the form of pinholes in the suction pipe can generally be stopped effectually by a thick coat of red-lead putty spread over the pipe where the leaks occur. This should be covered with several layers of canvas covered on both sides with red-lead putty and wound as tightly as possible around the pipe. The canvas should then be secured by wrapping it with strong twine or annealed copper wire, put on as tightly as possible. If the suction pipe is split, it is usually well to cover the split part with a piece of sheet metal, preferably sheet lead, bent to the curvature of the pipe and put on with red-lead putty. The canvas should be wrapped over this.

A permanent repair in case of pinholes can be made by drilling out the pinhole with a twist drill and tapping out

the hole. A closely fitting threaded plug of soft steel or wrought iron is then screwed in and the end riveted over.

89. Small pinholes in delivery pipes can often be stopped up by the same means given in Art. **88** for suction pipes. If the leak is extensive, however, it will generally be necessary to use a **pipe clamp.** Such clamps may be made in a good many different ways, according to the location and extent of the leak and the facilities for repair. One of the simplest pipe clamps is shown in Fig. 23. It consists simply of a piece of sheet iron or sheet steel of sufficient width to cover the leak and bent to the form shown. A piece of sheet packing, which may be covered with red-lead putty to advantage, is placed over the leak and the pipe clamp is then placed over this and the ends drawn together by the bolt shown.

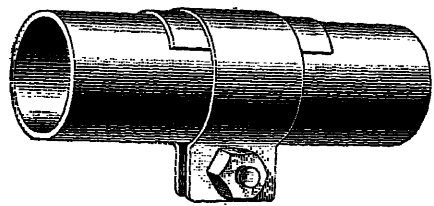


FIG. 23.

The clamp shown in Fig. 23 is only adapted for small pipes. For large pipes the clamp must be made in two halves.

90. Testing Air Chambers.—Air chambers must be absolutely tight. They are usually tested by closing all openings and then pumping air into them until the working pressure is reached, as shown by a pressure gauge. After 24 hours this gauge should show no reduction of pressure. If the air chamber does not pass this test, the leaks may be discovered by filling it with water subjected to the working pressure. If there are a number of leaks, the chamber should be condemned; if only a few small leaks exist, they can usually be effectually stopped by drilling a hole at the leak and screwing in a plug.

91. Leakage of Pistons and Plungers.—The plungers of inside-packed or center-packed plunger pumps should be

tight themselves, besides making a tight joint through the stuffingboxes, in order that water may not pass from one side to the other. The manner of testing will depend on their design, the general method of procedure being the subjecting of one side of the plunger to an air pressure or hydrostatic pressure at least equal to the working pressure. If leaks are discovered, judgment has to be used as to the manner of repairing them or whether to condemn the plunger. In some designs of inside-packed and center-packed pumps with closed hollow plungers, the weight of the plunger is so proportioned to its displacement as to relieve the stuffingboxes of nearly or quite all of its weight; it is then important that they be absolutely water-tight.

92. Leakage Past Pistons and Plungers.—With piston pumps and inside-packed plunger pumps there is liable to be considerable unnoticed leakage. If it is extensive, it can be heard by placing the ear against the pump chamber. It is best with this style of pump to make regular inspections for leakage past the plunger or piston, providing suitable pipes and apparatus by means of which pressure can be put on one side of the packing or piston while the other side is exposed for inspection. With outside-packed plungers there can be no unobserved leaks past the plungers, and this is the principal reason for their use.

93. Leaks Past the Valves.—Leaks past the suction and delivery valves can readily be tested when the piston or plunger is being tested for leaks past them. The delivery and suction valves should be tested separately; the fact that the column of water in the delivery pipe does not drain out while standing is not proof that both sets of valves are tight, since either set will support the water while the other set may be leaking badly.

94. To test the suction valves for leakage, disconnect the suction pipe or take any other convenient steps that will allow the leakage to be seen. Fill the delivery pipe full of water, having removed enough delivery valves to allow the pressure to reach all the suction valves, and observe which

valves, if any, are leaking. When there is a valve in the delivery pipe, this may be shut and water pumped into the pump cylinder with a small force pump, running the pressure up to the working pressure. Care must be taken, by removing delivery valves if necessary, that the pressure reaches all the suction valves.

95. The delivery valves can be tested by filling the delivery pipe or by closing the valve in the delivery pipe and pumping water into the delivery pipe between its valve and the pump delivery valve. The pump chamber must be open so that the leaks can be seen.

SURGING OF WATER IN PIPES.

96. By surging of the water flowing through pipes is meant that its velocity of flow not only is not constant, but that the direction of flow reverses for a short period. This condition often exists in pumping machinery having very long suction or delivery pipes. It may occur either in the suction pipes or in the delivery pipes, being, however, most severe in the latter. Crank-and-flywheel pumps, owing to the variation in the piston speed between the beginning and end of the stroke, are particularly liable to cause surging, which is due entirely to an irregular delivery.

97. Duplex direct-acting pumps, owing to the uniformity of delivery and the absence of heavy weights, such as flywheels, are little liable to cause surging, and when liquids must be moved through long mains, an instance of which are the long oil pipe lines, this pump is chosen. Crank-and-flywheel pumps forcing water through very high delivery pipes, as occurs in mine work, are seriously affected by the surging of the water. Air chambers do not help matters, but probably aggravate them by forming an elastic cushion for the column of water to rebound from. The effect of surging water is to vary the pressure on the pump and mains, sometimes from zero to twice the pressure due to the vertical height, resulting in broken pump chambers,

pipes, and not infrequently in damage to the working parts of the pump, for the actual resistance to these shocks is not met until they arrive at the flywheel rim.

98. The remedying of surging is not easy of attainment. Air chambers placed along the delivery pipe at intervals are employed occasionally, the aim being to break up the vibrations of the surging water and get them out of step or out of harmony with the motion of the pump. Alleviators are sometimes used in place of air chambers to relieve the shock, and not being so elastic do not encourage surging to the extent that air chambers do. When for economical reasons it is desired to use the crank-and-flywheel pump, the variations in pressure and the liability to surging can be very much reduced by using the three-throw crank with the pins set at 120° from one another.

99. Surging in long suction pipes is liable to occur especially when the water flows to the pump by gravity; this is not so difficult to overcome or so serious in its effects as surging in the delivery pipe, for the reason that the direction of the force resulting from the surge is through the pump valves and into the delivery, or in the natural direction of the water, while the shock due to surging in the delivery pipe comes against the valves and must be withstood by the machinery.

100. To prevent shocks due to surging reaching the machinery, a liberal sized air chamber is needed on the suction main near the pump, and in addition spring-loaded relief valves may also be fitted to the main. These relief valves simply limit the pressure due to an unusually heavy surge that cannot be taken care of by the air chamber.

PUMPING A MIXTURE OF WATER AND AIR.

101. In mine and artesian-well work, large quantities of air are often mixed with the water, due to local disturbances in the source of supply, such as water discharging into it in the form of spray. When such a mixture of air and water

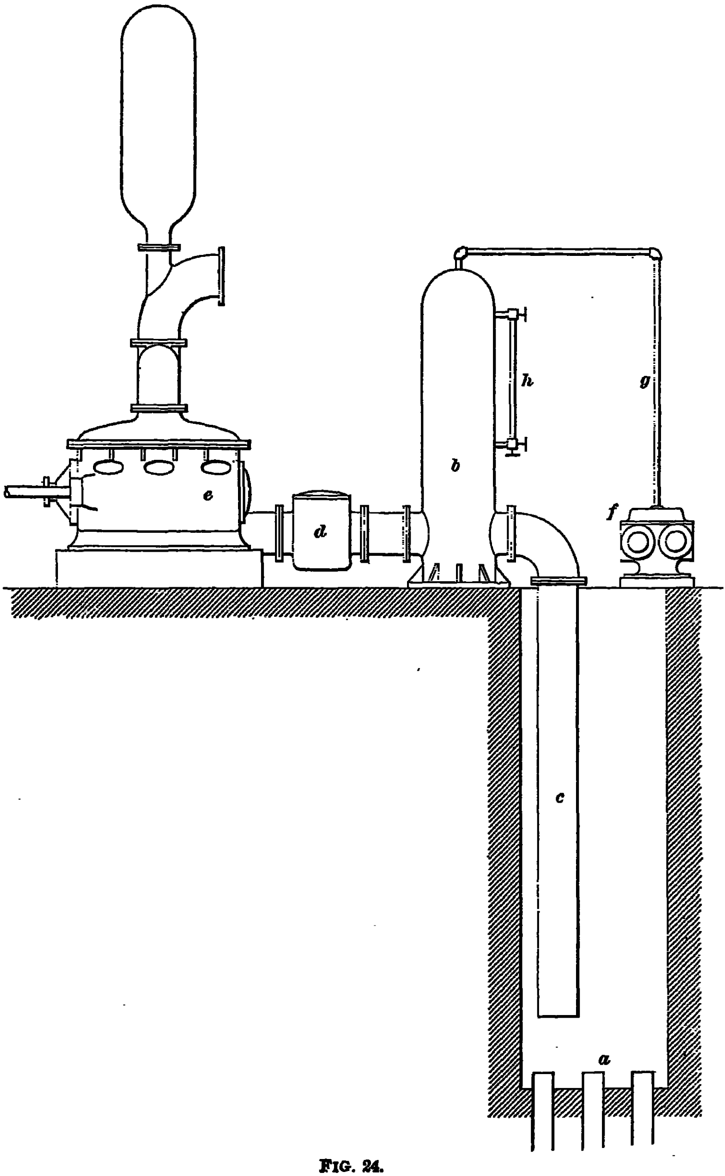


FIG. 24.

is pumped, the pump will have a jerky motion, that is, instead of moving steadily it will move in jumps, and in the case of direct-acting pumps there is danger of striking the cylinder heads. Besides, on account of the uneven discharge there will be violent disturbances in the delivery pipe. The only effectual remedy is to remove the air before it arrives at the pump.

102. Fig. 24 shows the installation of a pump taking its water from an artesian well *a*, the water being highly charged with air and gas. A large suction air chamber *b* is put into the suction pipe *c*; the water passes through the strainer *d* to the pump *e*. A vacuum pump *f* is connected by the pipe *g* to the top of the air chamber and not only maintains a vacuum in the chamber, but draws the air and gas out of the water in the chamber and before it reaches the pump. The gauge glass *h* not only shows the height of water in the air chamber, but also allows the bubbles of air and gas rising through the water to be seen. The vacuum pump is simply an ordinary steam pump pumping air instead of water; it is running constantly and its speed is regulated to suit the height of the water in the air chamber.

SETTING THE VALVES OF DUPLEX STEAM PUMPS.

103. The steam valves of duplex pumps have no outside or inside lap, consequently when in their central position they just cover the steam ports leading to opposite ends of the cylinders. With all these valves a certain amount of lost motion is provided between the jam nuts and the valve. This lost motion in small pumps is within the steam chest, while in large pumps it is outside and may be adjusted while the pump is in motion. The first move in the process of setting the valves of duplex pumps is to remove the steam-chest bonnets and to place the pistons in their mid-stroke position. To do this, open the drip cocks and move each piston by prying on the crosshead, but never on the valve lever, until it comes into contact with the cylinder head.

Make a mark on the piston rod at the steam-end stuffingbox gland. Move each piston back until it strikes the opposite head, and then make a second mark on the piston rod. Half way between the first and second mark make a third one. Then, if each piston is again moved until the last mark coincides with the face of the gland, the pistons will be exactly at their mid-stroke position. After placing the pistons in their mid-position, set the valves central over the ports. Adjust the locknuts so as to allow about $\frac{3}{16}$ inch lost motion on each side. The best way of testing the equal division of the lost motion is to move each valve each way until it strikes the nut or nuts and see if the port openings are equal. When the port opening has been equalized, the valves are set. The valve motion need not be and should not be disturbed while setting the valves. Too much lost motion will tend to lengthen the stroke and may cause the piston to strike the cylinder heads, while on the other hand when there is not enough lost motion, the stroke will be perceptibly shortened. The proper amount of lost motion to give a certain length of stroke can only be found by trial for each particular pump.

104. If only one valve of a duplex pump is to be set, bear in mind that it is operated by the piston of the opposite pump. Place that piston in its mid-position and then set the valve as previously explained.

PUMPS

(PART 3.)

CALCULATIONS RELATING TO PUMPS.

DISPLACEMENT.

1. The **displacement** of a pump for a single stroke is the volume of water that would be displaced (that is, driven out of the cylinder) by the piston or plunger during that stroke.

In calculating the displacement of a pump in a given time, care must be taken to consider the number of strokes during which water is discharged. Thus, for a single-acting pump, water is discharged only when the piston moves in one direction; and with the double-acting pump the number of strokes during which discharge occurs is equal to the total number of strokes that the piston makes. With a duplex double-acting pump, it is customary when giving the number of strokes per minute to refer only to the number of strokes made by one piston, which, obviously, is only one-half the total number of strokes made. As practice varies, however, among engineers in this respect, it is best to find out in each case, by inquiry, whether the number of strokes of one piston or of both pistons in a given time is meant when the number of strokes is given. In the case of a crank-driven pump, for a single single-acting pump the strokes will be equal to the revolutions of the crank; for a single

double-acting and a double single-acting crank-driven pump the strokes will equal twice the number of revolutions; for a triplex single-acting crank-driven pump the strokes will equal three times the number of revolutions; and for a triplex double-acting pump, six times the number of revolutions.

2. The displacement of a pump in a minute in cubic feet, gallons, or pounds is given by the following rule:

Rule 1.—*Multiply the length of stroke in inches by the mean effective area of the pump piston or plunger in square inches and the number of strokes per minute. The product is the displacement in cubic inches. To reduce the displacement to pounds, multiply by the weight of a cubic inch of the liquid pumped; to reduce to cubic feet, divide the displacement by 1,728; to reduce to Winchester gallons, divide the displacement by 231; to reduce to English imperial gallons, divide the displacement by 277.27.*

Or,

$$D_p = L\,A\,N\,S,$$

$$D_c = \frac{L\,A\,N}{1{,}728},$$

$$D_{ag} = \frac{L\,A\,N}{231},$$

$$D_{eg} = \frac{L\,A\,N}{277.27},$$

where

L = length of stroke in inches;
A = area of piston or plunger in square inches;
N = number of delivery strokes per minute;
S = weight in pounds of a cubic inch of the liquid,
D_p = displacement in pounds per minute;
D_c = displacement in cubic feet per minute;
D_{ag} = displacement in Winchester gallons per minute;
D_{eg} = displacement in English imperial gallons per minute.

3. Attention is here called to the fact that there are three different gallons in use, of which the Winchester, or wine, gallon, measuring 231 cubic inches, is most commonly

used in America. In Great Britain and her colonies the imperial gallon, holding 277.27 cubic inches, is largely used as a measure. In most English-speaking countries the beer or ale gallon of 282 cubic inches capacity is also used, but almost exclusively for measuring the liquids mentioned. When the discharge of a pump is given in gallons in the United States of America, it is always understood, unless distinctly stated otherwise, to be in gallons measuring 231 cubic inches.

4. The **mean effective area** of the piston or plunger is equal to the area corresponding to the diameter only in case of outside-packed plunger pumps. In case of inside-packed and center-packed plunger pumps and double-acting piston pumps, the mean effective area is found by dividing the sum of the piston or plunger area and the same area diminished by the area of the piston rod by 2. Thus, in a double-acting inside-packed plunger pump having a plunger 10 inches in diameter and a 2-inch piston rod, the mean effective area is $\frac{10^2 \times .7854 + (10^2 \times .7854 - 2^2 \times .7854)}{2}$ $= 76.97$ square inches. In case of a single-acting piston pump, which generally is a lift pump, the effective area will be the piston area; this should not be diminished by the area of the piston rod, although it is on the delivery side. In case of a differential pump having the plunger areas in the ratio of 1 to 2, the area of the smaller plunger is the effective area. In rough, approximate calculations of displacement, the correction for the area of the piston rod or plunger rod need not be made, and then the area of the piston or plunger is considered as the effective area. When the displacement requires to be accurately known, however, the mean effective area should be used.

EXAMPLE 1.—A single-acting plunger pump is driven by a crank whose radius is 8 inches and whose number of revolutions is 30 per minute. If the plunger is 6 inches in diameter, what is the displacement in cubic feet per minute?

SOLUTION.—The number of discharging strokes of the plunger is equal to the number of revolutions of the crank, or 30 per minute; the

length of the stroke is $8 \times 2 = 16$ inches. The area of the plunger is $6^2 \times .7854 = 28.27$ square inches. Applying rule **1**, we have

$$D_c = \frac{16 \times 28.27 \times 30}{1,728} = 7.85 \text{ cu. ft. per min.} \quad \text{Ans.}$$

EXAMPLE 2.—A center-packed double-acting duplex pump has plungers 24 inches diameter with 4-inch plunger rods. Each plunger makes 30 strokes per minute, the length of stroke being 32 inches. What is the displacement in American (Winchester) gallons per minute?

SOLUTION.—The mean effective area of the plungers is

$$\frac{24^2 \times .7854 + (24^2 \times .7854 - 4^2 \times .7854)}{2} = 446.1 \text{ square inches.}$$

Since the pump is duplex, there are $30 \times 2 = 60$ strokes per minute. Applying rule **1**, we get

$$D_{ag} = \frac{32 \times 446.1 \times 60}{231} = 3,707.8 \text{ gal. per min.} \quad \text{Ans.}$$

DISCHARGE.

5. The **theoretical discharge** of a pump is equal to the *displacement.*

6. The **actual discharge** is generally less than the displacement, owing to leakage past the valves and piston and also to the return of water through the valves while they are in the act of closing.

SLIP.

7. The difference between the displacement and the actual discharge, expressed as a percentage of the displacement, is called the **slip** of a pump.

8. Negative Slip.—When the column of water in the suction and discharge pipes of a pump is long and the lift moderate, the energy imparted by the piston during the discharge stroke may be sufficient to keep the column in motion during all or a part of the return stroke. Under these conditions, the actual discharge may be greater than the displacement, and the slip is then said to be *negative.*

Rule 2.—*To calculate the slip of a pump, find the difference between the displacement and the actual discharge, multiply it by 100, and divide the product by the displacement. The quotient will be the slip expressed in per cent. of the displacement.*

EXAMPLE.—A single-acting plunger pump with a plunger 8 inches in diameter and 36 inches stroke discharges 33.5 cubic feet of water per minute when making 35 discharging strokes. What is the slip?

SOLUTION.—By rule **1**, the displacement is

$$\frac{36 \times 8^2 \times .7854 \times 35}{1,728} = 36.652 \text{ cubic feet per minute.}$$

By rule **2**, the slip is

$$\frac{(36.652 - 33.5) \times 100}{36.652} = 8.6 \text{ per cent., nearly. Ans.}$$

WORK DONE BY A PUMP.

9. The **useful work** in foot-pounds done by a pump is the product of the water raised in pounds multiplied by the vertical distance in feet from the surface of the water in the well or supply reservoir to the outflow end of the discharge pipe.

10. The **actual work** is always greater than the useful work. Force is required to overcome the friction of the piston or plunger in the cylinder or stuffingbox, and considerable force is also required to overcome the friction of the water in its passage through the pipes and the valves and passages of the pump. Some force must also be expended in giving the water the velocity it has when it leaves the discharge pipe.

The theoretical force required to drive the piston is equal to its area multiplied by the pressure due to a head equal to the vertical distance from the surface of the water in the well to the outlet of the discharge pipe. The actual force can be found by the aid of a pressure gauge or indicator attached to the pump cylinder, which will give the actual pressure on the piston in pounds per square inch.

According to the principles of hydraulics and the flow of water through pipes, it is evident that the power required to overcome the frictional resistance of the water will be reduced by making the pipes large and direct and the passages through the valves and pump of ample size and as direct as possible, so as to avoid loss from sudden change of direction of flow.

HORSEPOWER OF PUMPS.

11. The indicated horsepower developed in the cylinder or cylinders of a steam-driven or compressed-air-driven pump is found in exactly the same manner as with a steam engine and from the same data. The horsepower usefully expended is given by dividing the useful work done by the pump in 1 minute by 33,000. The ratio of the usefully expended horsepower to the indicated horsepower is an indication of the mechanical efficiency of the pumping apparatus considered as a whole.

12. It is often required to estimate what horsepower will be required to pump a given quantity of water per minute to a given elevation or against a given pressure. This problem can only be solved approximately by a general rule, there being a number of variable factors entering into the solution, such as the general run and length of the piping, the design of the water end, the degree of workmanship, etc. The influence of some of these factors cannot be determined beforehand with any great degree of accuracy, and for that reason any general rule for estimating the required horsepower must be based on a low mechanical efficiency of the pumping apparatus in order to leave an ample margin for safety.

13. In estimating upon the probable horsepower, it is occasionally necessary to convert a given pressure into a head of water in feet that will exert the same pressure. This can be readily done by multiplying the given pressure by 2.3.

14. If the volume of water to be discharged per minute is given in cubic feet and the vertical height from the suction level to the discharge level in feet is known, the foot-pounds of work to be done is 62.5 × volume × vertical height, taking the weight of a cubic foot of water as 62.5 pounds. Consequently, the theoretical horsepower is

$$\frac{62.5 \times \text{volume} \times \text{vertical height}}{33{,}000},$$

or

$$\frac{\text{foot-pounds of work to be done}}{33{,}000}.$$

Assuming an efficiency of 70 per cent., the actual horsepower will be

$$\frac{100 \times \text{foot-pounds of work to be done}}{70 \times 33{,}000}$$

$$= \frac{\text{foot-pounds of work to be done}}{23{,}100}.$$

Hence the following rule:

Rule 3.—*To estimate the probable horsepower required to drive a pump, multiply the weight to be discharged per minute by the vertical lift and divide by 23,100.*

Or,

$$H_e = \frac{WL}{23{,}100},$$

where H_e = estimated horsepower;
W = weight of water discharged per minute in pounds;
L = vertical lift in feet.

EXAMPLE.—About what horsepower will be required to discharge 350 gallons of water per minute, the total lift being 320 feet?

SOLUTION.—The weight of the Winchester, or ordinary American, gallon is 8.34 pounds, nearly. Hence, the weight of water to be pumped per minute is $350 \times 8.34 = 2{,}919$ pounds.

Applying rule 3, we get

$$H_e = \frac{2{,}919 \times 320}{23{,}100} = 40 \text{ H. P., about.} \quad \text{Ans.}$$

15. When the weight of water to be discharged per minute and the pressure against which it is to be pumped are

known, the foot-pounds of work to be done is weight × pressure × 2.3. Assuming an efficiency of 70 per cent., the actual horsepower required is

$$\frac{100 \times \text{weight} \times \text{pressure} \times 2.3}{70 \times 33{,}000} = \frac{\text{weight} \times \text{pressure}}{10{,}043}.$$

Rule 4.—*To estimate the probable horsepower, multiply the weight of water to be pumped per minute by the pressure pumped against and divide by 10,043.*

Or, $$H_e = \frac{WP}{10{,}043},$$

where P = pressure per square inch;
W = weight of water per minute.

In rules 5, 6, 7, and 8 the letters have the same meaning as in rules 3 and 4.

Example.—A pump is to pump 400 cubic feet of water per hour against a pressure of 90 pounds per square inch. Estimate the probable horsepower required.

Solution.—Reducing the volume per hour to pounds per minute, we have

$$\frac{400 \times 62.5}{60} = 416.7, \text{ say } 417 \text{ pounds.}$$

Applying rule 4, we get

$$H_e = \frac{417 \times 90}{10{,}043} = 3.7 \text{ H. P., about. Ans.}$$

16. Rule 5.—*To estimate the vertical lift with a given horsepower, multiply the horsepower by 23,100 and divide by the weight of water to be delivered per minute.*

Or, $$L = \frac{23{,}100\, H_e}{W}.$$

Example.—A pump driven by a 10-horsepower engine is to discharge 2,000 pounds of water per minute. How high may this water be lifted, approximately?

Solution.—Applying rule 5, we get

$$L = \frac{23{,}100 \times 10}{2{,}000} = 115.5 \text{ ft. Ans.}$$

17. Rule 6.—*To estimate the probable discharge in pounds per minute, divide 23,100 times the horsepower by the vertical lift in feet.*

Or, $$W = \frac{23,100\,H_e}{L}.$$

EXAMPLE.—How many pounds of water per minute, approximately, can a pump driven by a 25-horsepower engine discharge at a height of 42 feet?

SOLUTION.—Applying rule **6**, we get

$$W = \frac{23,100 \times 25}{42} = 13,750 \text{ lb., about. Ans.}$$

18. Rule 7.—*To estimate the pressure that can be pumped against, multiply the horsepower by 10,043 and divide by the weight to be pumped per minute.*

Or, $$P = \frac{10,043\,H_e}{W}.$$

EXAMPLE.—A 9-horsepower pump is to discharge 6,000 pounds of water per minute. Estimate against what pressure this can be discharged.

SOLUTION.—Applying rule **7**, we get

$$P = \frac{10,043 \times 9}{6,000} = 15 \text{ lb. per sq. in. Ans.}$$

19. Rule 8.—*To estimate the probable discharge in pounds per minute, multiply the horsepower by 10,043 and divide by the pressure to be pumped against.*

Or, $$W = \frac{10,043\,H_e}{P}.$$

EXAMPLE.—How much water may a pump be estimated to discharge in Winchester gallons per minute when the pump is 40-horsepower and pumps against a pressure of 100 pounds per square inch?

SOLUTION.—Applying rule **8**, we get

$$W = \frac{10,043 \times 40}{100} = 4,017.2 \text{ pounds per minute.}$$

Since a Winchester gallon weighs 8.34 pounds, we have

$$\frac{4,017.2}{8.34} = 481.7 \text{ gal. per min. Ans.}$$

SIZE OF PISTONS AND PLUNGERS.

20. Before the size of a piston or plunger for the water end of a pump can be determined, the quantity of water to be pumped and the piston speed must be known. The piston speed is the number of feet traveled per minute by the plunger when *discharging* water; that is, it equals the length of the stroke in feet multiplied by the number of *working* strokes per minute. If the pump is double-acting, the number of working strokes is the same as the total number of plunger strokes, both forward and back; if single-acting, half that number. If the pump is duplex, it is advisable to consider only one side in determining the size of plunger or piston, designing it to suit one-half the total quantity of water to be delivered. In direct-acting steam pumps the piston speed is generally about 100 feet; at least it is customary to design them on this assumption, and then to run the pump faster or slower to suit the required delivery, opening or closing the throttle valve to vary the speed of the pump.

21. Knowing the actual volume of water to be discharged in 1 minute in cubic feet, the plunger or piston area in square feet will be $\frac{\text{discharge}}{\text{piston speed}}$, theoretically. But in practice the diameter of the plunger or piston is given in inches, hence the area should be expressed in square inches.

Then, $$\text{area} = \frac{\text{discharge in cubic feet} \times 144}{\text{piston speed in feet}},$$

and the corresponding diameter in inches will be

$$\sqrt{\frac{\text{discharge} \times 144}{.7854 \times \text{piston speed}}}.$$

22. Since there is always more or less slip of the water, it is usual to design the pump on the assumption that it must pump 1.25 times the actual amount of water. On this assumption the plunger or piston diameter in inches will be

$$\sqrt{\frac{\text{discharge} \times 1.25 \times 144}{.7854 \times \text{piston speed}}},$$

or
$$\sqrt{\frac{229 \times \text{discharge}}{\text{piston speed}}}.$$

Rule 9.—*To find the diameter of a plunger or piston in inches, multiply the discharge in cubic feet per minute by 229 and divide the product by the piston speed in feet per minute. Extract the square root of the quotient.*

Or,
$$d = \sqrt{\frac{229\,D}{S}},$$

where d = diameter of piston or plunger in inches;
D = actual discharge in cubic feet per minute;
S = piston speed.

When the discharge is given in pounds, gallons, or any other unit of volume, it should be reduced to cubic feet before applying rule **9**.

EXAMPLE.—What should be the diameter of a pump plunger required to discharge 130 Winchester gallons per minute, the speed of the plunger being 90 feet per minute?

SOLUTION.—Reducing the gallons to cubic feet, we have

$$\frac{130 \times 231}{1{,}728} = 17.378 \text{ cubic feet per minute.}$$

Applying rule **9**, we get

$$d = \sqrt{\frac{229 \times 17.378}{90}} = 6.65 \text{ in., nearly. Ans.}$$

23. Rule 10.—*To estimate the probable discharge in cubic feet, square the diameter of the plunger or piston in inches and multiply by the piston speed. Divide the product by 229.*

Or.
$$D = \frac{d^2\,S}{229}.$$

EXAMPLE.—How many pounds of water per hour may a duplex double-acting pump be expected to discharge when the diameter of the plungers is 6 inches, the length of stroke 24 inches, and each plunger makes 40 strokes per minute?

SOLUTION.—The piston speed is $\frac{24}{12} \times 40 = 80$ feet per minute. The probable discharge per minute in cubic feet, by rule **10**, is

$$D = \frac{6^2 \times 80}{229};$$

$$\text{per hour, } D = \frac{6^2 \times 80 \times 60}{229}.$$

The discharge in pounds per hour, taking 62.5 pounds as the weight of a cubic foot of water, is

$$D = \frac{6^2 \times 80 \times 60 \times 62.5}{229}$$

for one side of the pump. For both sides,

$$D = \frac{6^2 \times 80 \times 60 \times 62.5 \times 2}{229} = 94{,}323 \text{ lb. Ans.}$$

In applying rule **10** it is to be observed that the result will be less than given by multiplying the displacement per stroke by the number of strokes per minute, as called for by rule **1**. The reason for this discrepancy is obvious; rule **1** gives the theoretical discharge, while rule **10** gives about what the pump may actually be expected to discharge.

24. In direct-acting steam pumps the normal piston speed is generally 100 feet per minute. On this basis the probable discharge in cubic feet, by rule **10**, is $D = \frac{d^2 \times 100}{229}$, and in Winchester gallons the discharge is $\frac{d^2 \times 100 \times 1{,}728}{231 \times 229} = 3.26\,d^2$.

Rule 11.—*To roughly approximate the probable normal discharge of a direct-acting steam pump in gallons, multiply the square of the diameter of the plunger or piston by 3.26.*

Or, $$D_g = 3.26\,d^2,$$

where D_g = discharge in gallons per minute;
d = diameter of piston or plunger in inches.

25. The theoretical normal discharge in gallons per minute at a piston speed of 100 feet is given almost exactly by multiplying the square of the diameter of the plunger or piston by 4. For a duplex pump the discharge is double that given by rule **11**.

EXAMPLE.—What may the discharge in gallons of a duplex pump with 6-inch plungers be roughly estimated at?

SOLUTION.—Applying rule **11**, we get $D_g = 3.26 \times 6^2$ for each side, or

$$D_g = 3.26 \times 6^2 \times 2 = 235 \text{ gal. per min. Ans.}$$

26. Having determined the proper plunger or piston diameter for the chosen piston speed, it remains to choose either a length of stroke or a number of strokes in order to determine either the number of strokes or the length of stroke. The ratio of the diameter to the length of stroke varies between very wide limits in practice, being as low as 1 : 1 and as high as 1 : 5. Obviously, the greater the ratio, the fewer times will the valves have to be moved, hence a great ratio is generally chosen for pumps that have to run continuously in a hard, rough service. Having chosen a length of stroke, use the following rule:

Rule 12.—*To find the number of strokes, divide the piston speed in feet by the chosen length of stroke in feet. To find the length of stroke in feet, divide the piston speed in feet by the number of delivery strokes per minute.*

Or,
$$N = \frac{P}{L},$$

and
$$L = \frac{P}{N},$$

where P = piston speed;
N = number of delivery strokes per minute;
L = length of stroke in feet.

Example.—What should be the length of stroke for a piston speed of 100 feet if the number of strokes per minute is 40?

Solution.—Applying rule 12, we get

$$L = \frac{100}{40} = 2.5 \text{ ft.},$$

or
$$2.5 \times 12 = 30 \text{ in. Ans.}$$

SIZE OF STEAM END.

27. In a direct-acting steam pump the size of the steam-end cylinder depends on two factors, which are the steam pressure available and the resistance against which the pump is to force the water. The stroke of the steam piston and water piston obviously are the same, both being rigidly connected to the same rod.

28. The forces acting on the steam piston and water piston are equal when the area of the steam piston × the steam pressure = area of water piston × pressure pumped against. But in order that there may be an ample margin to overcome the frictional resistances, which make the actual resistance to the motion of the water piston greater and lessen the force that impels the steam piston forwards, the area of the steam piston should be, at least, 40 per cent. in excess of its theoretical area. On this basis, we have area of steam piston

$$= \frac{1.4 \times \text{area of water piston} \times \text{pressure}}{\text{steam pressure}},$$

and diameter of steam piston

$$= \sqrt{\frac{1.4 \times \text{area of water piston} \times \text{pressure}}{.7854 \times \text{steam pressure}}},$$

or diameter of steam piston

$$= \sqrt{\frac{1.8 \times \text{area of water piston} \times \text{pressure}}{\text{steam pressure}}}.$$

Rule 13.—*To find the minimum diameter of the steam piston of a direct-acting steam pump, multiply 1.8 times the area of the water piston in square inches by the pressure in pounds per square inch to be pumped against; divide by the available steam pressure and extract the square root of the quotient.*

Or, $$d_m = \sqrt{\frac{1.8\,a\,p}{P}},$$

where d_m = minimum diameter of steam piston in inches;
a = area of water piston;
p = pressure to be pumped against;
P = steam pressure available.

EXAMPLE.—What should be the minimum diameter of the steam piston for a pump having a plunger 8 inches in diameter, the available steam pressure being 75 pounds per square inch and the water to be pumped against a pressure of 200 pounds per square inch?

SOLUTION.—The area of the plunger is $8^2 \times .7854 = 50.27$ square inches. Applying now rule **13**, we get

$$d_m = \sqrt{\frac{1.8 \times 50.27 \times 200}{75}} = 15.5 \text{ in. Ans.}$$

29. It is to be observed that rule **13** applies equally well to steam- and air-driven pumps. It can also be applied to simple pumps of the crank-and-flywheel type using steam expansively. In the latter case, the mean effective pressure throughout the stroke must be taken as the available steam pressure. Rule **13** is especially useful in deciding whether a given pump will pump against a known pressure with the existing sizes of steam and water pistons. It will also be found very useful in selecting a pump for a given service from the catalogues of manufacturers.

30. In boiler-feed pumps the steam pressure available and the pressure pumped against are practically equal, so that it might be expected that the area of the steam piston would be made about 40 per cent. larger than the area of the water piston. In actual practice it is found, however, that pump manufacturers prefer to make the steam piston about 3 times the area of the water piston in very small pumps and about twice the area of the water piston in large pumps. The steam piston of boiler-feed pumps is made so largely in excess of what it really needs to be merely as a matter of safety; its large size simply tends to insure a prompt starting of the pump under almost all conditions likely to arise in practice.

31. The steam end of direct-acting pumps and of direct-connected crank-and-flywheel pumps, where the steam and water pistons move together, is rarely proportioned on the basis of horsepower required to do the work, it being much easier to calculate the size of the steam end by rule **13**.

32. When a power pump is driven by a separate steam engine, through the intervention of belting or gearing, the engine itself is generally selected on the basis of horsepower

required to do the work, and then the question as to what size of engine to use presents itself. This problem is capable of an infinite number of solutions, since a variation of either of the two factors—piston speed and mean effective pressure—will cause a difference in size. In general, the steam engine for driving a pump is selected in exactly the same manner, as far as its size is concerned, as a steam engine for any other service.

THE DUTY OF STEAM PUMPS.

DEFINITION.

33. The ratio between the work done by a pump and a certain amount of coal, steam, or heat units used to do the work is called the **duty** of the pump.

During a certain time, say an hour or a day, the pump will raise a quantity of water through a certain height and thus perform a definite amount of work. To do this work, the pump has received from the boilers a certain number of heat units or a number of pounds of steam; or, if the boilers are included as a part of the system, the work has been accomplished by consuming a certain amount of coal. The pump is credited with the work it has performed in the stated time and is charged with the number of heat units, pounds of steam, or pounds of coal it has used in doing the work. It is plain that the economy of the pump or pumping engine is measured by the ratio of the work performed to the steam consumed or the coal burned. Thus, if one pump does 50,000,000 foot-pounds of work with a coal consumption of 100 pounds and another under the same conditions does 36,000,000 foot-pounds and consumes only 60 pounds of coal, the latter is evidently the more economical, since the ratio of work to coal consumption is larger, being $(36,000,000 \div 60) \times 100 = 60,000,000$ foot-pounds of work with a coal consumption of 100 pounds.

DUTY BASED ON COAL CONSUMPTION.

34. When the duty is based on the consumption of coal, it is customary to assume 100 pounds of coal as the fuel unit; that is, the duty is defined as the number of foot-pounds of work performed for each 100 pounds of coal burned. Then,

$$\text{Duty} = \text{foot-pounds of work} \div \frac{\text{pounds of coal}}{100},$$

or,
$$\text{Duty} = \frac{\text{foot-pounds of work} \times 100}{\text{pounds of coal}}.$$

Rule 14.—*To find the duty of a pump per 100 pounds of coal, multiply together 100, the weight of water pumped in a given time in pounds, and the vertical distance in feet from the level of supply to the level of discharge. Divide the product by the coal consumption in the same time in pounds.*

Or,
$$D = \frac{100\,w\,h}{W},$$

where
$D =$ duty;
$w =$ weight of water in pounds;
$W =$ weight of coal in pounds;
$h =$ vertical lift in feet.

EXAMPLE.—A pump raises 130,000 pounds of water 60 feet and the operation requires the combustion of 25 pounds of coal. What is the duty?

SOLUTION.—Applying rule **14,** we have

$$D = \frac{100 \times 130{,}000 \times 60}{25} = 31{,}200{,}000 \text{ ft.-lb. per 100 lb. of coal. Ans.}$$

35. The duty based on the coal consumption is of practical value, as it gives an idea of the coal required by a pump of a given type for the performance of a stated quantity of work. It is clear, however, that if a comparison of the merits of two pumps is to be made, the coal must be of the same quality in each case. Further, the boilers supplying steam to the pumps should be of the same type or at least have the same evaporative capacity. This is a point of great importance. One hundred pounds of good bituminous or anthracite coal may, under favorable conditions, evaporate

1,000 to 1,100 pounds of water; that is, furnish that number of pounds of steam to the pump. In many cases, however, the 100 pounds of coal, if of inferior quality and burned under a poor boiler, will not furnish the pump more than 450 to 600 pounds of steam. Under such conditions the duty of the pump based on the coal consumption would not be a fair indication of its efficiency and would not serve as a satisfactory basis for comparing it with other pumps.

DUTY BASED ON STEAM CONSUMPTION.

36. In order to avoid the drawbacks incidental to basing the duty of pump on the coal consumption, it is the custom of some pump makers to specify that the coal consumption shall be estimated on the supposition that a pound of coal evaporates 10 pounds of water, or, in other words, furnishes 10 pounds of steam to the pump. To make this clear, suppose that in a duty trial 32,000 pounds of steam were used by the pump; the duty of the pump would be calculated on the assumption that the coal consumption was $32,000 \div 10 = 3,200$ pounds, though 5,000 pounds might actually have been used. If 1 pound of coal is assumed to furnish 10 pounds of steam, 100 pounds of coal will furnish 1,000 pounds of steam; hence, the duty based on steam consumption may be defined as the number of foot-pounds of work done by the pump per 1,000 pounds of dry steam supplied it. Then,

$$\text{Duty} = \frac{\text{foot-pounds of work} \times 1,000}{\text{pounds of steam}}.$$

Rule 15.—*To find the duty of a pump per 1,000 pounds of dry steam, multiply together the weight of water pumped in pounds, the vertical distance in feet from the level of supply to the level of discharge, and 1,000. Divide by the weight of steam supplied in pounds.*

Or, $$D = \frac{1,000\, w\, h}{S},$$

where S = weight of dry steam supplied in pounds and the other letters have the same meaning as in rule **14**.

EXAMPLE.—A pump lifted 7,920,000 pounds of water 126 feet with 8,100 pounds of steam. What is its duty?

SOLUTION.—Applying rule **15**, we get

$$D = \frac{1,000 \times 7,920,000 \times 126}{8,100}$$

$= 123,200,000$ ft.-lb. of work per 1,000 lb. of dry steam. Ans.

37. The basis of 1,000 pounds of dry steam is more scientific and better adapted for duty trials than that of 100 pounds of coal, but it is open, nevertheless, to objections. Not only is it difficult to determine the exact weight of dry steam entering the pump, but also 1,000 pounds of steam at 160 pounds pressure will do more work in the cylinder than 1,000 pounds of steam at 60 pounds pressure. If scientific accuracy is sought, the pressure of the steam should be specified in addition to the weight.

DUTY BASED ON HEAT UNITS SUPPLIED.

38. On account of the objections to the basis of comparison then used, a committee of The American Society of Mechanical Engineers in 1891 recommended a new basis for the estimation of duty. Whether the furnace consumes 100 or 200 pounds of coal, whether the steam is at 60 or 160 pounds pressure, wet or dry, the steam cylinders of the pump or pumping engine receive in a given time a definite number of British thermal units. We have seen that if each of two pumps is allowed 100 pounds of coal to do a certain amount of work, one of the pumps may be at a disadvantage on account of the poor quality of the coal or the inefficiency of the boiler. If each is allowed 1,000 pounds of dry steam, there may be an inequality because of a difference in the steam pressure in the two cases. If, however, each pump is furnished with an equal number of heat units, each has exactly the same stock in trade, and the merit of each pump can be gauged by the use it makes of the heat units furnished it, that is, by the ratio of the work performed to number of heat units supplied.

39. If a pound of water has a temperature of 212°, it requires 966.1 B. T. U. to change it to steam at atmospheric pressure. If the water has originally a lower temperature or is converted into steam at higher pressure, more B. T. U. are required to accomplish the change. Roughly speaking, if the temperature of the feed and pressure of the steam are not given, about 1,000 to 1,100 B. T. U. are equivalent to a pound of steam. Therefore, 1,000 pounds of steam are equivalent to about $1,000 \times 1,000 = 1,000,000$ B. T. U.

40. Looking at the question in another light, a pound of good coal when burned produces about 13,500 to 14,000 B. T. U. by the combustion. A boiler of fairly good efficiency will utilize perhaps 10,000 of these 13,500 B. T. U., the rest being lost by radiation, in the production of chimney draft, and in other ways. From 100 pounds of coal the boiler is able to extract $100 \times 10,000 = 1,000,000$ B. T. U., which are eventually given up to the pump. It thus appears that 100 pounds of coal and 1,000 pounds of steam are each approximately equivalent to 1,000,000 B. T. U.; for this reason, the committee of The American Society of Mechanical Engineers recommended that the new basis for estimating duty should be 1,000,000 B. T. U.

41. The heat-unit basis is now very extensively used and is recommended in preference to the others. It may be expressed as follows:

The duty of a pumping engine is equal to the total number of foot-pounds of work actually done by the pump divided by the total number of heat units in the steam used by the pump, and this quotient multiplied by 1,000,000. The heat units in the steam used for driving the auxiliary machinery, such as the air pump and circulating pump of the condenser, if one is used, and the boiler-feed pumps are charged as heat units supplied to the pump.

42. The number of foot-pounds of work done by the pump is to be found as follows: A pressure gauge is attached to the discharge pipe and a vacuum gauge to the

suction pipe, both as near the pump as convenient; then the net pressure against which the pump plunger works is equal to the sum or difference in the pressures shown by these two gauges increased by the hydrostatic pressure due to the difference in level of the points in the pipes to which they are attached. In case the gauge in the suction pipe indicates a vacuum, the sum of the pressures indicated by the gauges is taken, but when the water flows into the suction pipe under a head, so that the suction gauge indicates a pressure above the atmospheric pressure, the difference in the two pressures indicated by the gauges is taken.

43. The number of foot-pounds of work done during the trial is equal to the continued product of the net area of the plunger in square inches (making allowance for piston rods), the length of the plunger stroke in feet, the number of plunger strokes made during the trial, and the net pressure in pounds per square inch against which the plungers work.

44. The pressure corresponding to the vacuum in inches indicated by the gauge on the suction pipe is found by multiplying the gauge reading in inches by .4914, and the pressure corresponding to the difference in the level of the two gauges by multiplying this difference in feet by .434. The number of heat units furnished to the pump is the number of British thermal units in the steam from the boilers and is to be determined by an evaporation test of the boilers.

Rule 16.—*To determine the duty of a pump per 1,000,000 B. T. U., multiply the net pressure against which the plunger works, in pounds per square inch, by the net area of the plunger in square inches, by the average length of stroke in feet, the total number of delivery strokes made during the trial, and by 1,000,000. Divide the product by the total number of B. T. U. supplied during the trial.*

Or, $$D = \frac{1{,}000{,}000\,(P \pm p + S)\,A\,L\,N}{H},$$

where D = duty;

P = pressure in pounds per square inch in the discharge pipe;

p = pressure in pounds per square inch in the suction pipe, to be added in case of a vacuum and to be subtracted in case of pressure above atmospheric pressure in the suction pipe;

S = pressure in pounds per square inch corresponding to difference in level between the gauges;

A = average effective area of plunger in square inches;

L = length of stroke of pump plunger in feet;

N = total number of delivery strokes;

H = total number of B. T. U. supplied.

EXAMPLE.—A crank-and-flywheel pump has two double-acting water plungers, each 20 inches in diameter and 36 inches stroke. Each plunger has a piston rod 3 inches in diameter extending through one pump-cylinder head.

During a 10-hour duty trial the total heat in the steam supplied to the engine was 35,752,340 B. T. U. and the engine made 9,527 revolutions. If the average pressure indicated by a gauge on the discharge pipe was 95½ pounds, the average vacuum indicated by a gauge on the suction pipe 8¼ inches, and the difference in level between the centers of the vacuum and the pressure gauge 8 feet, what was the duty?

SOLUTION.—The area of a plunger 20 inches in diameter is 314.16 square inches and the cross-sectional area of a rod 3 inches in diameter is 7.07 square inches. Since the rod extends through only one end of the pump cylinder, the average effective area of the two ends of each plunger is $314.16 - \frac{7.07}{2} = 310.63$ square inches.

The pressure corresponding to a vacuum of 8¼ inches is $p = 8.25 \times .4914 = 4.05$ pounds per square inch, and the pressure corresponding to a difference in level of 8 feet is $S = 8 \times .434 = 3.47$ pounds per square inch.

Since there are two double-acting plungers, the total number of plunger strokes corresponding to 9,527 revolutions is $N = 9,527 \times 4 = 38,108$.

Applying rule **16**, we get

$$D = \frac{1,000,000 \times (95.5 + 4.05 + 3.47) \times 310.63 \times 3 \times 38,108}{35,752,340}$$

$$= 102,328,800 \text{ ft.-lb. per } 1,000,000 \text{ B. T. U.} \quad \text{Ans.}$$

DUTY BASED ON VOLUME OR WEIGHT PUMPED.

45. In large pumping plants it often happens that the pressure pumped against is either constant or practically so. In such a plant a record is often kept for the purpose of comparing the performance of the plant from week to week or month to month with its former performances. The records may be kept in number of gallons pumped per pound of coal; in cubic feet pumped per pound of coal; in weight of water in pounds or tons pumped per pound of coal; or the record may be kept per ton or bushel of coal, etc. Duty computed on such a basis is spoken of as **gallon duty, cubic-foot duty, pound duty, ton duty,** etc.; while such a duty is very valuable in showing variations in efficiency of a given plant at different times, it cannot be used as a basis of comparison between the performances of different pumping plants, and when so used will be utterly misleading.

Instead of keeping the records in terms of quantity of water pumped per pound of coal, they may advantageously be kept in terms of water pumped per dollar; the records then show variations in efficiency in their true light.

EXPRESSING THE DUTY OF A PUMP.

46. The question "in what terms shall the duty of a pump be expressed" depends for its answer on the purpose for which it is required that the duty be known. If the duty is merely required to be known in order that the performances of a given pump at different times may be compared with one another, the duty may be based on coal consumption, steam consumption, or volume pumped per some unit of fuel or money. If, however, the performance of a pump is to be compared with that of others working probably under entirely different conditions, the foot-pounds of work done per 1,000,000 B. T. U. is the only true basis of comparison.

AVERAGE DUTIES.

47. Small direct-acting pumps for general service have a duty of 15,000,000 foot-pounds per 1,000 pounds of steam used. Compound direct-acting pumps of 5,000,000 gallons capacity in 24 hours should give a duty of 50,000,000 foot-pounds per 1,000 pounds of steam used. Large municipal pumping engines of 20,000,000 gallons capacity in 24 hours have given a duty of 160,000,000 foot-pounds per 1,000 pounds of dry steam used by the engine.

Centrifugal and rotary pumps have a duty depending on the type of engine used to drive them, and since they usually run at high speed and the conditions for economical performance are good, an economical type of engine can be used and the duty of the combined unit thus made to compare very favorably with that of the reciprocating pump.

48. Tests of the duty of pumps and pumping engines have generally been made when the machinery was in first-class condition. It is customary to run these machines from 6 months to 1 year after they are installed before making the test, the object being to bring all the journals into a good bearing condition; also, the piston and all the other rubbing surfaces will be much improved by the polishing and the working of oil into the pores of the iron during running. These high duties can only be maintained by the closest attention to every detail by the operating engineer. Indicator cards should be taken from both the steam and the water ends of the pumps every week and closely compared with previous indications to see that the highest state of efficiency is being maintained within the working parts of the pump.

EFFICIENCY OF VARIOUS TYPES OF PUMPS.

49. When the **efficiency of a pump** is spoken of, its *mechanical* efficiency is generally meant, unless stated otherwise. This is measured by dividing the actual or net horsepower of the machine by its indicated horsepower, and the quotient, when multiplied by 100, will be the efficiency

expressed in per cent. Very small direct-acting steam pumps have an efficiency of about 50 per cent., the efficiency increasing with the size of the pump up to about 80 per cent. The efficiency of direct-acting steam pumps and also of pumps in general increases with the size by reason of the decrease in the ratio that the frictional resistances bear to the indicated horsepower as the size of the pipes and passages is increased. The reason that the frictional resistances decrease can readily be seen when it is considered that by doubling the diameter of a pipe and keeping the velocity of flow the same, the discharge will be increased four times, while the surface that the water is rubbing against is only doubled.

50. Large vertical municipal pumping engines have shown an efficiency as high as 96 per cent.; horizontal medium-size crank-and-flywheel pumps show efficiencies as high as 90 per cent. The efficiency of centrifugal, rotary, and screw pumps varies between 40 and 66 per cent., about, depending on the size; small pumps are less efficient than larger ones. This efficiency of centrifugal, rotary, and screw pumps is the *efficiency of the pump itself*, and not the combined efficiency of the pump and engine, or motor, driving it.

SIZE OF SUCTION AND DELIVERY PIPES.

51. Experience has demonstrated that for satisfactory work the flow of water in the suction pipes of pumps should not exceed 200 feet per minute, and it should not be more than 500 feet in the delivery pipe for a duplex double-acting pump, or 400 feet for a single-cylinder double-acting pump.

Knowing the volume of water that is to flow through or to be discharged from a pipe in 1 minute, the area of the suction and delivery pipes can readily be determined.

The volume of water in cubic feet discharged from a pipe in 1 minute is equal to the velocity in feet per minute times the area of the pipe in square feet. Then,

$$\text{the area of the pipe} = \frac{\text{volume in cubic feet per minute}}{\text{velocity in feet per minute}}.$$

As there are 144 square inches in a square foot,

$$\text{the area of the pipe in square inches} = \frac{144 \times \text{volume in cubic feet per minute}}{\text{velocity in feet per minute}}.$$

Rule 17.—*To find the area of a pipe in square inches to discharge a given volume of water per minute, divide the product of the volume in cubic feet and 144 by the allowable velocity in feet per minute.*

Or, $$A = \frac{144\ V}{v},$$

where A = area of pipe in square inches;
V = volume to be discharged per minute;
v = allowable velocity.

When the weight of water is given in pounds, divide it by 62.5 to reduce it to cubic feet; when the volume is given in Winchester gallons, divide it by 7.48 to reduce it to cubic feet.

EXAMPLE.—What should be the areas of the suction and delivery pipes for a single double-acting pump that is to discharge 6,250 pounds of water per minute?

SOLUTION.—Reducing the weight to cubic feet, we have $\frac{6{,}250}{62.5} = 100$ cubic feet. Then, applying rule **17**, we have

$$A = \frac{144 \times 100}{200} = 72 \text{ square inches}$$

as the area of the suction pipe, and

$$A = \frac{144 \times 100}{400} = 36 \text{ square inches}$$

as the area of the delivery pipe. The nearest standard nominal sizes of pipe to be used would be 10-inch and 7-inch. Ans.

52. The velocity with which water will flow through the delivery pipe of a pump when the area of the water cylinder, the area of the delivery pipe, and the piston speed of the pump are known, is given by the following rule:

Rule 18.—*Multiply the area of the water piston by the piston speed and divide this product by the area of the delivery pipe.*

Or,
$$v = \frac{a S}{A},$$

where v = velocity in feet per minute;
A = area of delivery pipe in square inches;
a = area of water piston in square inches;
S = piston speed in feet per minute.

EXAMPLE.—If the water piston of a pump has an area of 12 square inches and moves at a speed of 100 feet per minute, what will be the velocity of the water in the delivery pipe if the latter has an area of 2 square inches?

SOLUTION.—Applying rule **18**, we get

$$v = \frac{12 \times 100}{2} = 600 \text{ ft. per min.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. The plungers of a center-packed double-acting duplex pump are 20 inches in diameter and the plunger rods are $3\frac{1}{2}$ inches in diameter. Each plunger makes 45 strokes per minute, the length of stroke being 24 inches. What is the displacement in cubic feet per minute?
Ans. 386.69 cu. ft.

2. In the above example, if the pump delivers but 360 cubic feet per minute, what is the slip? Ans. 6.9 per cent.

3. Approximately, what horsepower will be required to deliver 60 cubic feet of water per minute, the total lift being 470 feet?
Ans. 76.3 H. P.

4. What is the probable horsepower required to deliver 3,500 gallons of water per hour against a pressure of 115 pounds per square inch?
Ans. 5.57 H. P.

5. A pump driven by a 25-horsepower engine is to discharge 60 cubic feet of water per minute. How high may this water be lifted, approximately? Ans. 154 ft.

6. Approximately, how many gallons of water per hour can a pump driven by a 30-horsepower engine deliver at a height of 65 feet?
Ans. 76,701.7 gallons.

7. Approximately, against what pressure can a 20-horsepower pump discharge 2,500 cubic feet of water per hour?
Ans. 77 lb. per sq. in.

8. About how many cubic feet of water per minute may a 75-horse-power pump be expected to discharge against a pressure of 150 pounds per square inch? Ans. 80.3 cu. ft. per min.

9. A pump is required to discharge 1,800 cubic feet of water per hour. If the speed of the plunger is 100 feet per minute, what should be the diameter of the plunger? Ans. 8.29 in., nearly.

10. If the plunger of a double-acting pump is 10 inches in diameter and the length of stroke is 24 inches, how many gallons of water per hour may the pump be expected to deliver if it makes 45 strokes per minute? Ans. 17,639 gal. per hr.

11. Roughly estimate the discharge in gallons of a direct-acting steam pump having a plunger 7 inches in diameter.

Ans. 159.7 gal. per min.

12. If the piston speed is 90 feet per minute and the length of stroke 2 feet, how many strokes per minute will the pump make? Ans. 45.

13. Calculate the minimum diameter of the steam piston for a pump having a plunger 12 inches in diameter, the pressure to be pumped against being 175 pounds per square inch and the available steam pressure 100 pounds per square inch. Ans. 18.87 in.

14. What is the duty per 100 pounds of coal of a pump that raises 330,000 pounds of water 125 feet and requires 110 pounds of coal to perform the operation? Ans. 37,500,000 ft.-lb.

15. If 20,016 pounds of steam are consumed by a pump in lifting 1,200,000 gallons of water 150 feet, what is the duty per 1,000 pounds of dry steam? Ans. 75,000,000 ft.-lb.

16. A double-acting pump has a stroke of 40 inches; the diameter of the plunger is 24 inches and the diameter of the piston rod, which extends through one pump-cylinder head, is 3½ inches. During a 12-hour duty trial the total heat supplied to the engine was 47,652,500 B. T. U. and the engine made 23,200 strokes. What was the duty of the pump per 1,000,000 B. T. U. if the average pressure indicated by the gauge on the discharge pipe was 122 pounds, the average vacuum indicated by a gauge on the suction pipe 5 inches, and the difference in level between the centers of the vacuum gauge and pressure gauge was 10 feet?

Ans. 93,555,123 ft.-lb.

17. Calculate the area of the suction and delivery pipes for a single-acting pump that is to discharge 1,250 gallons of water per minute.

Ans. $\begin{cases}\text{Suction pipe, 120.3 sq. in.}\\ \text{Delivery pipe, 60.15 sq. in.}\end{cases}$

18. If the plunger of a single-acting pump has a speed of 85 feet per minute and a diameter of 6 inches, what will be the velocity of the water in the delivery pipe if the latter has an area of 6 square inches?

Ans. 400.5 ft. per min.

SELECTION OF PUMPS.

SERVICE OF DIFFERENT TYPES OF PUMPS.

53. Introduction.—The service for which a pump is required determines its general type, that is, whether it is to be a plunger pump, a rotary pump, a centrifugal pump, or a screw pump.

54. Reciprocating Pumps.—The various types of reciprocating pumps are selected when high efficiency is required and a fluid for which they are suited is to be pumped.

55. Rotary Pumps.—The rotary pump is chosen when the fluid to be pumped is water holding in suspension large masses of soft material. It is much used in paper mills for pumping the pulp from one stage of its manufacture to another. Rotary pumps are small and occupy, relatively, but little space for their capacity; they are also light, simple, and inexpensive, but are low in efficiency and are short lived, particularly if the material pumped contains much sand or other grit. The rotary pump is used with good success on some steam fire-engines, where light weight and simplicity are more important than high efficiency.

56. Centrifugal Pumps.—Centrifugal pumps are used where large volumes of water are to be lifted to moderate heights. They are also well adapted for pumping large quantities of dirty water, and, hence, are also much used for dredging and for sewage pumping. The efficiency of the centrifugal pump is low, but it is extremely simple and occupies comparatively little space for its capacity. Like the rotary pump, it has no valves and the flow is continuous. It is less affected by sand and grit than is the rotary pump. Neither the rotary pump nor the centrifugal pump requires much, if any, foundation.

57. Displacement Pumps.—Under the head of displacement pumps may be classed the pulsometer, which has

no running parts. This type of pump is well adapted for pumping all kinds of gritty water and is used for sinking and contractor's purposes. It is very simple in construction, low in first cost, and is not liable to get out of order. The class of pumps known as air lifts are principally used for artesian-well service; they require an air compressor for operation, but the apparatus itself is simple and low in first cost.

58. Screw Pumps.—Screw pumps are adapted for the handling of thick liquids, such as hot tar, pitch, paraffin, soap, etc. They have a uniform discharge and occupy small space; a much higher efficiency is claimed for them than for rotary or centrifugal pumps.

RECIPROCATING PUMPS.

CLASSIFICATION.

59. The reciprocating pump is, in general, the most efficient and hence the most common pump. It is built in a large variety of designs to suit different conditions and varies in size between very wide limits. Reciprocating pumps may be classified in accordance with the service for which they are intended as boiler-feed pumps, general-service pumps, tank or light-service pumps, fire pumps, low steam-pressure pumps, pressure pumps, mine pumps, sinking pumps, ballast pumps, wrecking pumps, deep-well pumps, sewage pumps, vacuum pumps, power pumps, municipal pumping engines, etc.

BOILER-FEED PUMPS.

60. Boiler-feed pumps are used for supplying steam boilers with their necessary water supply. For low pressures they are usually made of the piston pattern or the inside-packed plunger patterns. The cylinders are generally brass lined; the valves are brass or hard composition, with

composition springs and guards, and the pumps, hence, are suitable for handling hot water. For pressures above 135 pounds the outside-packed plunger type is preferred. Boiler-feed pumps are made both vertical and horizontal and for pressures from 50 pounds to 300 pounds per square inch. They vary in size from those having water plungers 1 inch in diameter to those having plungers 10 inches in diameter. The single-cylinder type is much used for boiler feeding, but, perhaps undeservedly, they have not the reputation for continuous action under all circumstances that is given to the duplex pump. Power pumps are often used for boiler feeding.

61. Whenever possible the boiler feeding apparatus should be in duplicate, so that the stoppage of one set will not affect the running of the plant. This end is generally secured by installing both a pump and an injector, each having a capacity sufficient for the needs of the plant.

62. Steam-driven crank-and-flywheel pumps are occasionally used, but they are open to the serious objection that they cannot always be run slow enough to suit the demand without stopping on the centers. In very large electrical installations, the electrically driven power pump is the most economical and satisfactory arrangement. Mills and factories often use the two-throw power pump having a movable crankpin, by means of which the stroke and hence the quantity of water pumped can be adjusted to suit the requirements. By this means a constant supply of feedwater equal to the demands for steam can be obtained, which is superior to the practice of pumping large quantities of water into the boilers at intervals. Boiler-feed pumps should not be required to run faster than 100 feet per minute piston speed. The velocity of water through the suction pipe should not exceed 200 feet and through the delivery should not be more than 400 feet. If the pipes are long or fitted with elbows, the velocity should be correspondingly decreased.

63. In determining the proper capacity of a pump for boiler feeding, the pump should be selected in reference to the amount of steam the boilers must supply. This is rarely only the amount used by the engine; in fact, in many industrial establishments much more steam is needed for other machinery than for the engine. Hence, it is best to always base the estimate as to the amount of water required on the maximum capacity of the boilers.

64. The maximum water consumption may be estimated in pounds per minute by one of the following rules, which hold good for average practice under natural draft. It will be observed that no rule based on the so-called "boiler horsepower" is given, for the reason that this is too variable a quantity to place any reliance on.

Rule 19.—*For plain cylindrical boilers multiply the product of the length and diameter in feet by .18.*

Rule 20.—*For tubular boilers multiply the heating surface in square feet by .06.*

Rule 21.—*Multiply the grate surface in square feet by 1.7.*

Rule 22.—*Multiply the estimated coal consumption in pounds per hour by .17.*

65. Whenever possible the feed-pumps should be located in the boiler room, so as to be directly in sight and in charge of the boiler attendant. In very large installations it is common to arrange the pumps in a separate pump house, they being then in charge of one of the assistant engineers, the boiler attendants regulating the supply to each battery by valves in the feedpipes.

GENERAL-SERVICE PUMPS.

66. General-service pumps are a line of pumps placed on the market by many of the pump builders to be used for any service where the water pressure does not exceed 150 pounds. They are generally of the plunger type and are built in sizes varying from those having a 4-inch to those

having a 16-inch plunger, and of a capacity varying from 100 gallons to 2,500 gallons per minute. They may be used for any service such as boiler feeding, fire, hydraulic elevator, or anywhere where the pressure to be pumped against is not greater than the limit stated.

TANK OR LIGHT-SERVICE PUMPS.

67. Tank or light-service pumps are of the same general form and interior construction as general-service pumps, except that the plungers are much larger in proportion to the steam cylinders, equalling or exceeding them in diameter. Such pumps cannot be used to feed their own boilers, but they are sometimes fitted with an attached pump for this purpose. Light-service pumps are commonly built of the same capacity as general-service pumps, but can only pump against low pressures.

FIRE PUMPS.

68. Fire pumps are most frequently of the duplex double-acting type with a ratio of area of steam cylinder to water piston of about 4 to 1. The duplex engine is chosen for this service on account of its simplicity and the peculiar adaptability of its motion to the high speed that is sometimes required in this service. A fire pump is frequently fitted up with a number of nozzles for hose connection. It should have relief valves, air and vacuum chambers of large capacity, steam and water gauges, priming pipes, and all the necessary valves.

Fire pumps, as implied by the name, are intended for use in case of fire, and are required to throw a large volume of water at high pressure.

LOW-PRESSURE STEAM PUMPS.

69. Low-pressure steam pumps are pumps intended for localities where only a low steam pressure is available, as in apartment houses, public and private buildings, etc.,

in which the pressure at which the steam heating system is worked does not exceed 5 to 10 pounds per square inch. The ratio of cylinder areas is about 9 to 1, the steam cylinder being the larger. Otherwise they are fitted up similar to pumps for general service. In some cases a hand power attachment is provided so that the pump can be worked by hand when the steam pressure is down.

PRESSURE PUMPS.

70. Pressure pumps are designed especially for use in connection with hydraulic lifts, cranes, cotton presses, testing machines, hydraulic machine tools of all kinds, and hydraulic presses, also for oil pipe lines, mining purposes, and such services as require the delivery of liquids under very heavy pressure. These pumps are invariably of the outside-packed plunger type and generally have four single-acting plungers working in the ends of the water cylinders, the latter having a central partition. The water valves are contained in small chambers capable of resisting very heavy pressures and ingeniously arranged for ready access. All materials used in the construction of the water end must be first class and suitable to the pressure used, which ranges from 750 pounds per square inch to 1,500 pounds per square inch. The water ends of these pumps are frequently made of hard, close-grained composition for medium pressures, and of steel castings for the heavier pressures.

MINE PUMPS.

71. Perhaps no other class of pump requires as much experience and skill to select as the mine pump. The reason for this is the wide variations in service, conditions of operation, head or pressure to be worked against, and the destructive nature of the water to be pumped. Nearly all the pumps at present installed are placed entirely below the surface. In former times the Cornish, or bull, pump was the favorite, but it is today abandoned for the more compact

and less expensive modern mine pump. The water end of the modern high-pressure mine pump may be described as having outside-packed plungers; strong circular valve pots independent of one another, but bolted to the working chamber, to the suction and delivery pipes, and to one another. Frequently the whole inside of the water end of the pump, from the suction nozzle to the discharge flange, is lined with wood, lead, or some other acid-resisting substance. Sometimes the entire water end is made of an acid-resisting bronze. Unless the service is light the outside-packed plunger pump is recommended for mine work; the valves should be preferably metallic valves in separate pots or chambers. Whether the pump shall be simple, compound, or triple expansion depends much on the price of fuel. In the anthracite coal regions the compound mine pump is now very common for sizes as small as 1,000,000 gallons in 24 hours, and they are invariably compounded for larger sizes, while the triple-expansion direct-acting pump is found in several of the mines.

72. Compound crank-and-flywheel high-duty pumps using the steam expansively have but recently been installed in the coal mines; in the iron and copper mines, where the cost of fuel is very high, the highest types of pumping engines have long been used.

73. When the larger types of high-duty pumps are used, the mine workings are generally so arranged that all the water runs to one large basin or sump near which a chamber of sufficient size is cut to contain the pump, which is surrounded and protected by suitable devices to maintain it in a high state of efficiency.

74. In many mines, strength and simplicity are the controlling elements in selection, for the reasons that many mines are compelled to use a large number of medium sized pumps and, for commercial reasons, use only one man whose business it is to make the rounds of the various pumps, giving each one but a few minutes' attention in a day. They generally have to stand rough usage, and the water pumped

is of such a corrosive quality that repeated renewals of parts of the water end are absolutely necessary. After heavy rains or other causes of flooding, the pumps are often required to run for days completely submerged and must pump both themselves and the mine dry. It can be readily seen that a pump for such service must be strong, simple, ready of access, and all of its parts of such construction that they can be readily taken apart or renewed.

SINKING PUMPS.

75. Sinking pumps are used in sinking or deepening mine shafts. There is little choice in their selection; generally speaking, they should be simple, strong, and capable of working in any position. The valves should be of the simplest possible construction and accessible for renewal with a minimum of labor and time. The valve motion should be simple and protected from dirt and drippings. They are regularly built single cylinder and duplex and are steam or electrically driven. With electric sinking pumps the protection of the electrical parts must be very complete.

BALLAST AND WRECKING PUMPS.

76. Ballast and wrecking pumps are principally confined to the marine service. The ballast pump is used on steamers having an extensive system of water ballast; also, for handling petroleum in bulk on oil-tank steamers. It is distinctively a special pump. The wrecking pump has a somewhat wider sphere. As its name implies, it is used principally by wrecking companies on the Atlantic and Pacific coasts and along the Lakes and is constructed with particular reference to reliability, portability, and general efficiency. It is well adapted to other services requiring the delivery of large volumes of water within the range of lift by suction. It has no forcing power, the water being merely delivered over the top of the pump, and it is single-acting, the water piston being fitted with valves. It is a

very light form of pump in proportion to the work it will do, is simple, durable, and not liable to derangement or breakage. It is also well adapted to drainage and irrigating purposes.

DEEP-WELL PUMPS.

77. Deep-well pumps, like sinking pumps, give little field for choice except in the pump-driving mechanism, which is as varied as the agent available to operate them, the principal agents being steam, electricity, gas, and windmills. The pump is usually a lifting pump having a bucket packed with numerous hydraulic leathers and working within the casing; it is usually given a very long stroke. These pumps do not handle gritty water successfully. Probably the best practical solution of the deep-well pump problem will be found in the air lift, which in principle and operation is quite simple.

SEWAGE PUMPS.

78. Sewage pumps are built in various types. When the lift is low, which condition is most common in sewage disposal, the centrifugal pump is the cheapest to install, but when economy and efficiency are important factors, the centrifugal pump must give place to the more expensive but more efficient reciprocating pump. Probably the largest single pumping engine ever constructed is the sewage pump for the city of Boston, which has a capacity of 70,000,000 gallons in 24 hours.

79. It will readily be seen that the selection of a type of sewage engine will depend much on the capacity of the installation and the price of fuel delivered at the station. The principal characteristics of the sewage engine are in the valves, which must be provided with very large ports to allow fairly large objects to pass through the pump without obstructing its valves. The valves are frequently made in the form of large leather-faced doors or flap valves, giving

nearly the full area of the pipe. The sewage pump does not differ in other respects from pumps for general service.

POWER PUMPS.

80. **Power pumps** are among the oldest styles of pumps, and may be developed by driving any type of reciprocating pump by other means than the use of a directly attached steam, gas, or air cylinder. Power pumps are very often geared or belted and with the increasing application of electricity the electric power pump is coming into more extensive use.

81. Power pumps may be used for any service and are frequently found in municipal water works, being often driven by a turbine or a Pelton waterwheel. In large electric-lighting, heat, and power plants, the power pump is much used for boiler feeding; in this case the pumps are usually triplex, giving a steady flow of water, and are driven by electric motors, the current being furnished by the main generators. This is probably the most efficient and economical boiler feeder that has been developed.

82. The power pump is used quite extensively in the mines. An electric motor being the driver, the system admits of many various sized pumps being placed at the different sumps throughout the mines and driven by one large and economical generating unit at the surface.

83. The selection of a power pump in preference to other types depends on conditions that, to some extent, may be gathered from the above applications; the choice, however, depends much on the kind of power available to run the machine. Where water-power is available, either for gearing directly to the pump or for generating electric current to drive the pump at a remote distance, the power pump may advantageously be chosen. It should be remembered in this connection that a steam pump should be installed to take

care of the feedwater when the main engines are stopped and no current is available for driving the power pump.

84. In private houses, hotels, office and public buildings the electric-power pump is a favorite, and to avoid the noise of gears the reduction in speed is made by friction drives of various types; rawhide gearing is also used to some extent. The construction of the water end of power pumps does not differ from other pump constructions for the same service.

MUNICIPAL PUMPING ENGINES.

85. While the **municipal pumping engine** may be of any size and capacity, and while some of the pumps already discussed, as the general-service and power pump, may be, and are, frequently used, the term usually implies the highest type of pumping engine that can be constructed as regards economy and efficiency. The refinement is more exacting as the capacity of the pump increases. For small municipal pumping engines, say of 2,000,000 to 5,000,000 gallons capacity in 24 hours, the compound and triple-expansion direct-acting engine is used, the degree of expansion depending on the price of fuel and the capital available for the investment. For installations of from 5,000,000 to 20,000,000 gallons capacity, the high-duty direct-acting engine, that is, the direct-acting engine with high-duty attachment, and the crank-and-flywheel engine are rivals for the installation; while for large municipal pumping engines above 20,000,000 gallons capacity in 24 hours, the vertical triple-expansion condensing three-crank single-acting, or differential, plunger beam type may be said to have no equal. With the latter type of engine a duty of 160,000,000 foot-pounds of work per 1,000 pounds of steam used by the engine is now common. Steam pressures of 175 pounds are common, while the number of expansions are as high as 22 to 26, and every reasonable device known in the art of steam engineering is used to the end of breaking records in securing a high duty.

VACUUM PUMPS.

86. Vacuum pumps are chiefly used in connection with jet condensers and siphon condensers. A vacuum pump is in reality an air pump, it being used for pumping air out of closed vessels. There are two general types of vacuum pumps, which are **dry vacuum pumps,** or pumps that handle air only, and **wet vacuum pumps,** or pumps that handle both air and water. Vacuum pumps are also used in some manufacturing operations where a high degree of vacuum is required, being used in connection with the vacuum pans found in sugar houses, with glycerin pans, etc.

RELATIVE MERITS OF DIRECT-ACTING AND CRANK-AND-FLYWHEEL PUMPS.

87. The relative merits of the two types of machine for a particular size, other conditions being equal, are such that it is a very difficult matter to decide which type is superior. For pumping small quantities of water, say up to 700 gallons per minute, and in localities where coal is not expensive, the direct-acting pump, either simple or compound, should prove a good investment. The objection to the direct-acting pump for large sizes is its waste of steam as compared with the crank-and-flywheel pump; it has an additional objection that is sometimes argued against it, which is *short-stroking*. This defect reduces its economical performance in that it requires some steam to fill up the space due to the incomplete stroke, but since the incomplete stroke is due to too high a compression, the compressed steam must have nearly filled the space before fresh steam was admitted, so that the loss is not so very great after all. Short-stroking reduces the capacity of the machine somewhat. In the common types of direct-acting pumps, the steam is not worked expansively; in compound and triple-expansion pumps, some degree of expansion is obtained, usually a little more than the ratio of low-pressure cylinder to high-pressure cylinder. By making the reciprocating parts

heavy and running the pump at some fixed minimum speed, an early cut-off can be effected in the high-pressure cylinder, the balance of the stroke being completed by the inertia of the reciprocating parts; in this way an increased degree of expansion is possible.

Another method of securing a considerable degree of expansion in the direct-acting pump is by means of the high-duty attachment. With the same degree of safety the speed of the direct-acting pump is very much less than is possible with the crank-and-flywheel pump. The direct-acting pump in which any attempt is made at economy will occupy quite as much space as the crank-and-flywheel pump of the same capacity, but the direct-acting pump is lower in first cost.

88. Probably the most objectionable feature of the crank-and-flywheel pump, which is an inherent one, is that the velocity of discharge varies throughout the stroke. This is due to the fact that while the flywheel rotates at a uniform speed, the pistons and plungers move with a variable speed, varying from zero at the beginning of the stroke to the maximum speed near mid-stroke and then decreasing to zero at the end of the stroke. This variation in velocity produces shocks, and hence requires the water end of a flywheel pump to be of heavier construction than a similar end for a direct-acting pump. The valve area of a flywheel pump requires to be considerably larger than for a direct-acting pump, not only because of its capacity for higher speeds, but also because the velocity of the plunger, when the connecting-rod is at right angles to the crank arm, is somewhat in excess of 1.57 times the mean velocity of the plunger. In addition to the greater valve area and strength required in flywheel pumps, it is necessary to use some means to reduce the shocks to the mechanism and parts of the pump. This is accomplished by providing large air chambers, preferably one over each deck for high pressures; for very high pressures and long columns of water, an alleviator is necessary.

89. The main advantage of the crank-and-flywheel pump is its economy, which, in turn, is due to the fact that the

steam may be expanded to any permissible degree; it also readily admits of all the refinements known of securing high-duty performance, and with a proper arrangement of details, it can be made quite as safe as ordinary machines. For extreme high duties the crank-and-flywheel pump is always chosen, and to reduce the shocks due to a variable discharge a favorite type is the three-crank machine. The combined delivery from three plungers is tolerably uniform and the arrangement readily lends itself to the extremely economical triple-expansion condensing engine.

90. The crank-and-flywheel engine is more expensive than the direct-acting machine, and when high degrees of expansion are used occupies considerably more room. It is generally more complicated, but is more accessible, except in such cases as where an effort is made to minimize space, when by making the engine back-acting it is liable to become quite inaccessible.

91. The piston speed of direct-acting pumps rarely exceeds 100 feet per minute, while the piston speed of crank-and-flywheel pumps is commonly 300 feet and sometimes 400 feet. With pumps of the controlled-valve type, piston speeds of 560 feet are reached. This difference in the piston speed of the direct-acting and crank-and-flywheel pumps shows that they must be compared on the basis of water delivered rather than on the relative size of similar parts.

92. Even for very small sizes, the crank pump is sure in its action and is not liable to get out of order; this cannot be claimed for some of the single-cylinder direct-acting pumps having steam-thrown valves. The crank pump is limited as to its slowest speed, however, since the speed must be sufficient to store energy enough in the flywheel to carry the crank over the dead centers. This objection can be overcome to a great extent by using the by-pass, which allows part of the water to be returned to the suction, thus decreasing the work on the pump.